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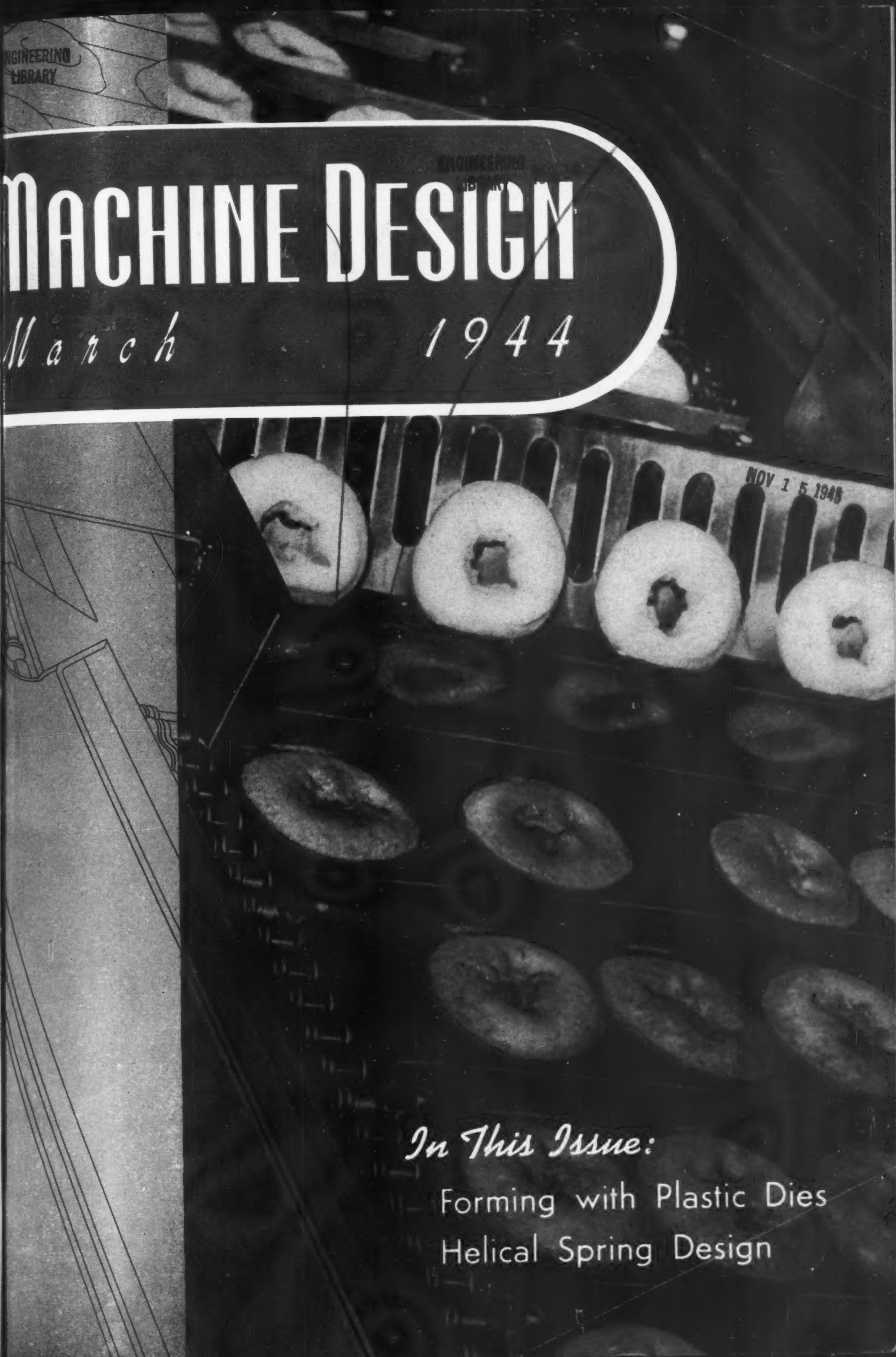
MACHINE DESIGN

March 1944

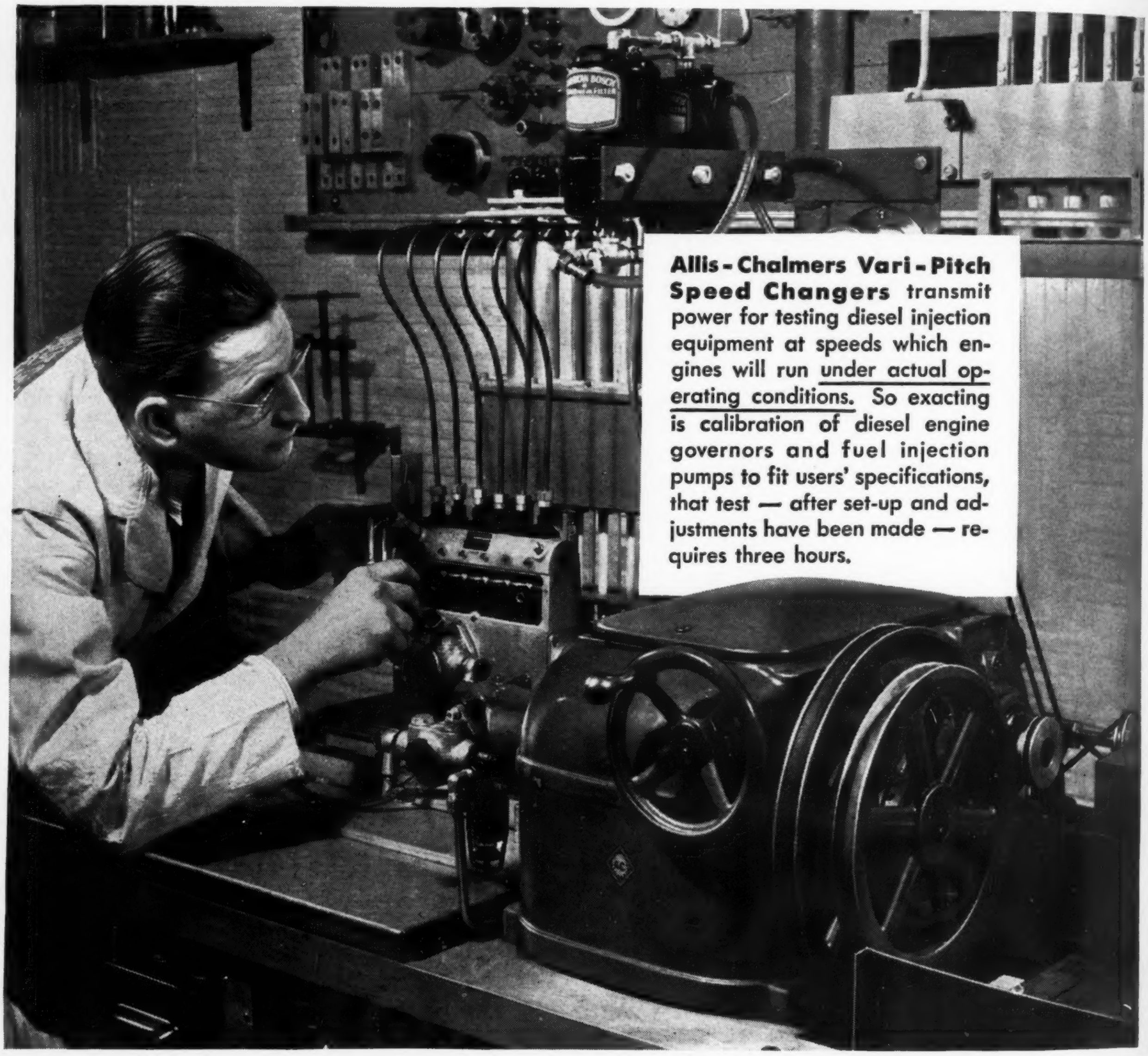


In This Issue:

Forming with Plastic Dies

Helical Spring Design

PRODUCT OF THE MONTH



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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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Itemized Index

Classified for Convenience when Studying Specific Design Problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

This issue at a glance

From 200,000 to 13,000! That's the Startling Reduction . . . .

. . . . in manhours achieved by an American manufacturer in producing the 2000th as compared to the first assembly-line four-engine bomber. Many factors contribute to this heartening accomplishment. Not the least of these is the speed with which forming dies can be produced—and when need be, modified—when made of cast plastic. John Delmonte's "How Plastic Tooling Speeds Production", Page 99, presents the latest data on this modern forming process.

Recent Findings in Spring Design . . . .

. . . . have made possible the fulfillment of rigid, high-precision requirements of spring applications in modern mechanisms of war. Pertinent factors—many times overlooked—which influence the design of springs, are discussed by Dr. Wahl in an article based on a chapter from his forthcoming book *Mechanical Springs*. Page 107.

About 3,000,000 Rivets Go Into the Structure . . . .

. . . . of a large modern airplane. Fastening the myriad parts used in the airframes of such planes constitutes fully half their total cost. Thus the development of rational methods for joint design—with an eye cocked toward more economical production—takes on timely significance. What actually happens in a loaded joint? The facts don't jibe with traditional conceptions. Page 131.

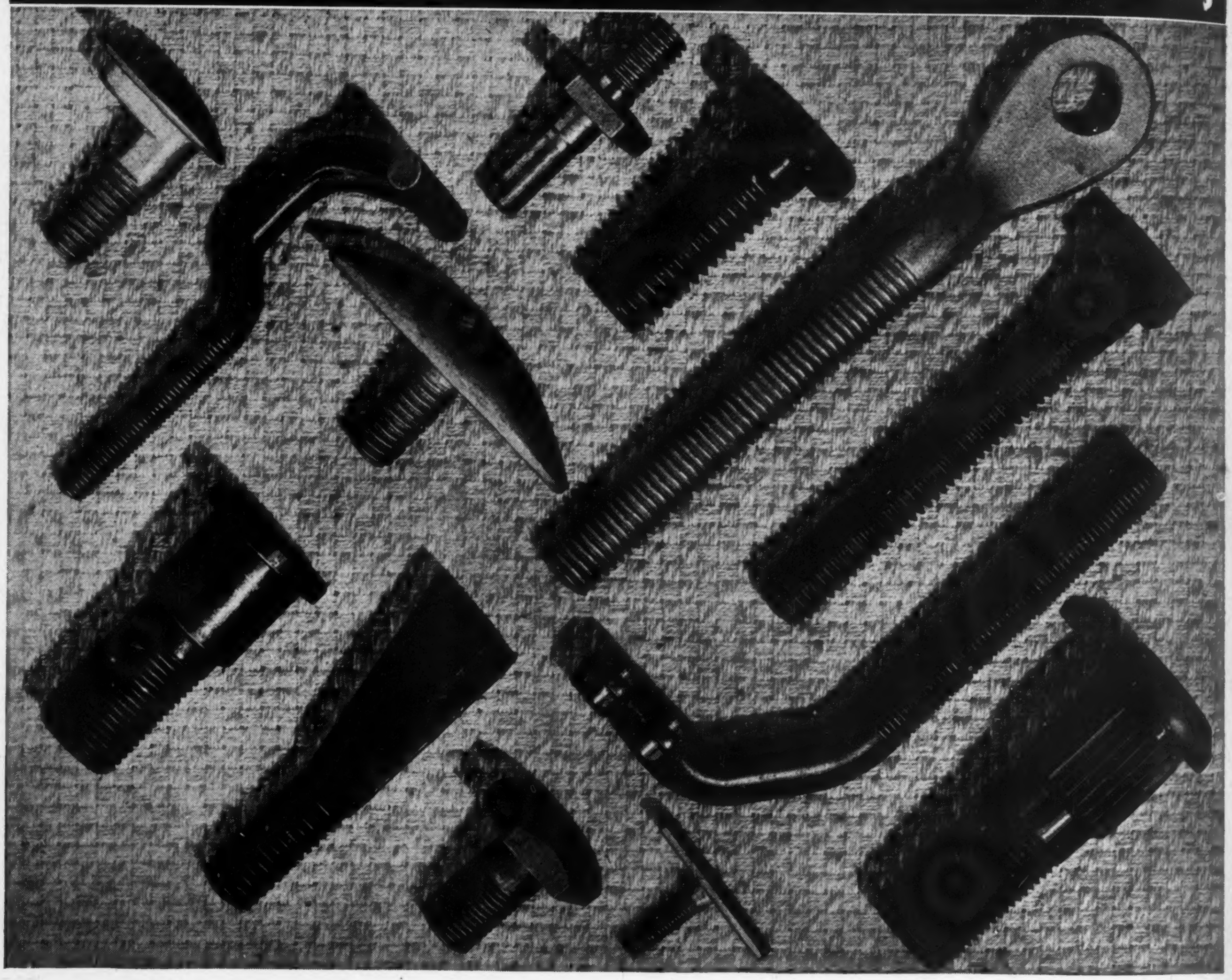
Will Postwar Private Planes Depart Drastically . . . .

. . . . from those of prewar days? What will they offer in speed, safety, comfort and maneuverability? How much will they cost? You'll want to read William D. Hall's "Forecasting the Private Plane of the Future". It's realistic, based on factual thinking—not wishful dreaming. Page 124.

Government Restrictions on the Use of Metals . . . .

. . . . for nameplates have recently brought plastics into this field more strongly than ever before. John W. Greve discusses the factors influencing design and application of plastic nameplates, as they are used specifically on machinery, in his interesting article "Selecting Plastic Nameplates". Page 112.

Making strong the things that make America strong



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Make a reminder now that cold-forging may be an answer to the greater strength or lower cost requirements of the product you plan for post-war development.

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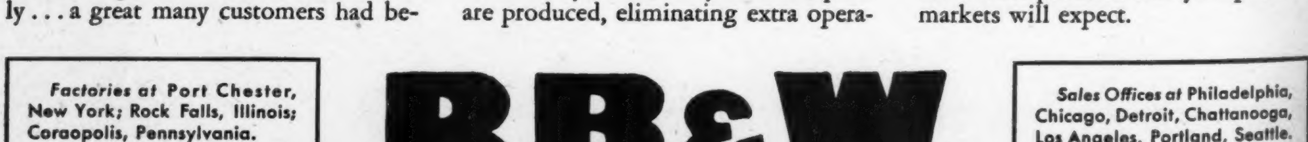

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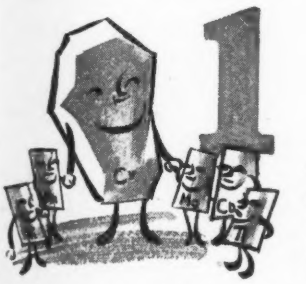
Basic Facts about

STAINLESS STEELS

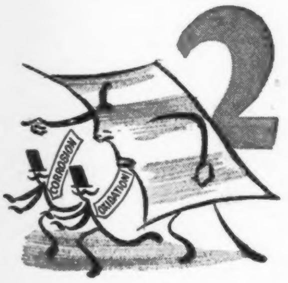
A STAINLESS STEEL PRIMER



STAINLESS STEELS

are corrosion-resistant steels containing at least 12 per cent chromium with or without other alloying elements, such as nickel, manganese, molybdenum, columbium or titanium. They are supplied in a wide range of analyses.

The stainless steels vary in corrosion resistance, workability, wear resistance, and physical properties according to the amount of chromium, carbon and other modifying elements present.



VALUABLE PROPERTIES INCLUDE:

Good corrosion and oxidation resistance . . . varying almost in direct proportion to the amount of chromium present.

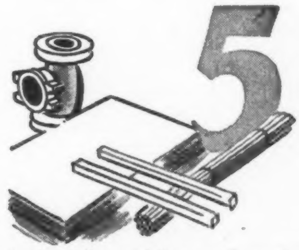
High strength-weight ratio in some cold-rolled types permitting fabrication of strong, light-weight trains, aircraft, and other structures.

MAINTENANCE

of stainless steels is simple. Washing with soap, water, and a cleanser recommended for stainless steel will keep the surface bright and free from surface deposits.

FABRICATION

is accomplished by almost all common methods. They can be machined, spun, deep-drawn, forged, punched, stamped, and otherwise mechanically-worked. They can be welded by all the common welding methods. If the austenitic steels are stabilized with columbium or titanium, and columbium-bearing welding rod is used, no annealing is necessary after welding.



MANY FORMS

are available in the common analyses including sheet, plate, strip, tubing—both seamless and welded—bars, wire, cable, welding rod, and a variety of cold-rolled shapes. Stainless steel is also supplied as foundry castings.

APPLICATIONS

of stainless steel are numerous. Because of their resistance to corrosion and oxidation, as well as their high strength and bright surface, stainless steels have been used in hospitals, chemical plants, oil refineries, railroad trains, aviation equipment, and power plants. In the present emergency, they are available only for those industries participating in war production.

• • •

Although we do not make steel, we have for more than 35 years produced "Electromet" ferro-alloys and metals used in making steel. With the knowledge accumulated from this experience, we are in a position to give impartial advice. If you have a specific problem concerning the manufacture, fabrication, or use of stainless or other alloy steels, consult us without obligation.

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Topics

PLASTIC TOOLING, one of the nation's war-born developments adaptable to postwar uses, has been given impetus by the formation of the National Society of Plastic Tooling by representatives of five major aircraft firms. Design possibilities of this method of tooling are discussed in John Delmonte's article in this issue.

NEW ENGINE MOUNTS which require less rubber have been announced by Goodyear. For Flying Fortresses, the mounts allow the engines to reach their vibration peak much earlier, permitting more accurate aiming of bomb sights and guns, longer life for each Fortress and less nerve strain on the crew members.

ANODIC FILMS for magnesium alloys make them as noncorrosive and abrasion resistant as aluminum alloys. Weight increase for .04-inch thick magnesium sheet treated on both sides is only .8 per cent or 14 per cent of the weight of a prime coat of paint and two coats of lacquer. The new film is described as extremely tight and resistant to electrical current, readily withstanding the potential of 110 volts. Post-war factors for use of this coating include its high degree of finish and the possibilities of introducing dies into the anodic solution to produce a wide variety of colors such as have been applied to aluminum.

SPECIAL PLATING for copper or brass parts in aircraft instruments which carry high-frequency current has been developed. Because of the skin effect in parts carrying high frequencies, it is necessary that the plating have high surface conductivity. Also, high corrosion resistance is needed. Nickel and gold are not used because of the high resistance of the former and the high price of the latter. The answer to the problem proved to be the use of a plating technique which utilizes anodes of copper, tin and zinc alloy.

FIRST Army Ordnance Distinguished Service Award has been presented to the Society of Automotive Engineers "in recognition of outstanding and meritorious engineering advisory services in war and peace; in design, manufacture and maintenance of Ordnance materiel." The award specifically recognizes the extensive automotive engineering advisory services rendered to the Ordnance Department by S.A.E. committees created after World War I and, more recently, by groups operating under the S.A.E. war engineering program established more than a year before Pearl Harbor.

SERVO-MECHANISM principles that enable tanks to fire on the run, keeping the gun on the target despite deviations of the vehicle, might be applied to high-speed trains and other vehicles to give "floating rides". Calculations, according to Westinghouse engineers, show that the power requirement to stabilize vertical movements of a railroad coach is only about three horsepower. The mechanism necessary to accomplish this is small enough to fit into an overnight bag.

CHEMICAL TREATMENT endows wood with plastic properties by converting the wood cellulose into resin. Developed by duPont, the process utilizes resin-forming chemicals and makes the product dimensionally stable and hard.

RADIO-FREQUENCY CURRENT for generating heat in raw-material preforms makes possible faster production of molded plastic parts without the usual risk of damage to molds and with marked improvement in quality. According to RCA engineers it is felt that the higher curing efficiency of electronic heating methods may make practicable the use of less costly plastic compounds where a high degree of strength is not an especially important factor.

DIELECTRIC HEATING and induction heating for drying and fusing porcelain enamels are two processes being studied by manufacturers which may have revolutionary effects upon the application of porcelain finishes. Electrolytic cleaning of the metal base as well as electrolytic precipitation of enamels in spraying are also being studied and are expected to produce better bonds together with savings in materials.

CONDUCTIVE RUBBER gun-heating pads on machine guns and aerial cannon give American aircraft another margin of safety added to their performance. These pads, developed by U. S. Rubber Co., preheat gun breeches while the plane is grounded or maintain required temperatures for instantaneous use at high altitudes. The conductive rubber eliminates sparks and static electricity, and until this use was found for the material it had been employed principally to reduce the danger of explosion in war plants where highly flammable products are manufactured.

MACHINE DESIGN

How Plastic Tooling Speeds Production

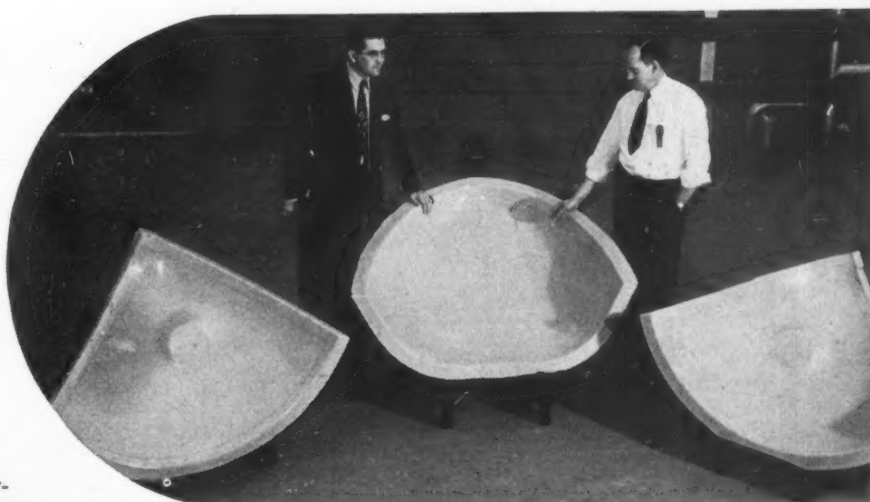
By John Delmonte
Technical Director
Plastics Industries Technical Institute

TWO years have passed during which industry has had an opportunity to experiment with and apply plastic materials to some of their tooling problems. Most active in this work have been aircraft manufacturers faced with limited production and the need for speed in turning out finished pieces. Knowledge of these activities can be helpful to machine designers who find that plastic tooling can augment their design program. Many lessons have been learned, and from applications—successful and otherwise—the following facts emerge:

1. Under no conditions do plastic tools possess the strength or mechanical durability of metal tools; hence plastic tooling will serve only limited production.

2. Plastic tools can, under proper conditions, be produced at an appreciably lower cost (one-third the cost of metal in some instances) and in much shorter periods of time. Designers now will find it practical to prepare experimental models with the aid of plastic tools and jigs.

3. Some types of plastic tools are now being used in regular production in the forming of metal parts. One of the large cast phenolic dies used at the Lockheed Aircraft Corp. appears in *Fig.* 1.



Photo, courtesy Lockheed Aircraft Corp.

Fig. 1—Made at a fraction of the cost of previous methods and in much less time, these phenolic dies are used in regular production of metal parts

4. Cast plastic tools are successfully adapted to low-stressed form or contour blocks, drill jigs, clamps for irregular-shaped castings. An example of this type of tooling appears in *Fig.* 2, which illustrates a wing-root fairing drill fixture for the P-38 "Lightning".

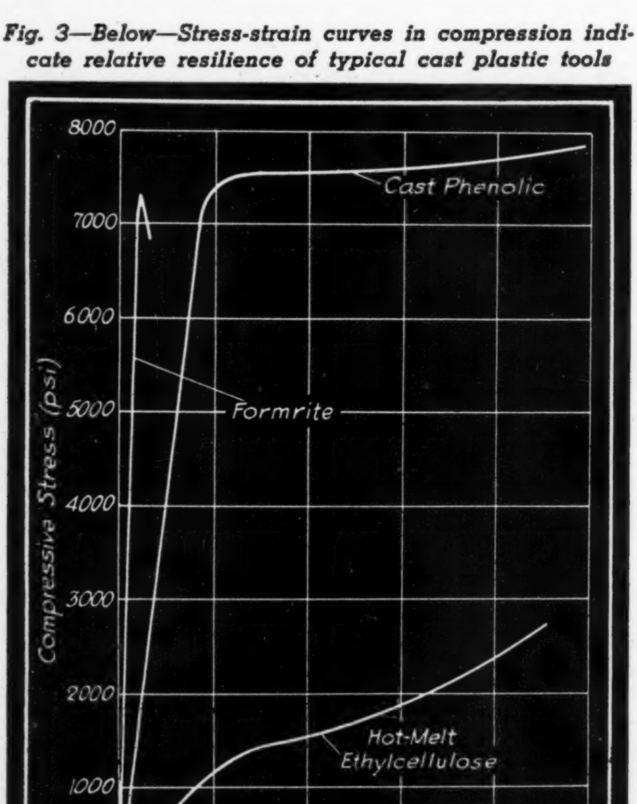
5. Flat plates, bolsters and spacer blocks are made from sheet forms of plastics.

Developments in nonmetallic tooling found origin in applications of plaster of paris forms and patterns. While ideal from a cost and fabrication standpoint, plaster lacked sufficient strength for most tools and jigs. With improved, higher-strength plaster of paris such as Hy-

Photo, courtesy Lockheed Aircraft Corp.

Fig. 2—Above—Cast plastic form and metal part for wing-root fairing drill fixture. Part is used on P-38

Fig. 3—Below—Stress-strain curves in compression indicate relative resilience of typical cast plastic tools



drocal and Hydrastone there was some sacrifice in ease of workability.

Cast phenol-formaldehyde tools began to make their appearance during 1941 in industrial plants in Los Angeles, and activities with these materials have been on the upgrade ever since. In addition to the organic plastic tooling developments, considerable progress was achieved with certain inorganic cements, such as magnesium oxychlorides which came into use in competition with the organic plastics.

Of more recent origin are the developments in hot-melt ethylcellulose compounds—Plastalloy and Thermocast—which have proved of special value in drop-hammer dies for forming metal parts. They are formulated to be somewhat resilient or rubber-like, a special feature which allows them to yield without cracking during the drawing of metal. No allowance is made for metal thickness in casting these dies, a fact which should be a boon to machine designers experimenting with different metal thicknesses for their parts.

In Table I is a useful comparison of the various castable nonmetallic materials employed in tooling problems. Many of the materials are inferior in tensile qualities (generally under 1500 pounds per square inch ultimate), hence could not be considered in drawing operations on sheet metal, for example, unless reinforced within a retaining ring. Likewise only the hot-melt thermoplastics have adequate impact resistance for drop-hammer work. Most of the materials possess sufficient compression strength (7000 to 10,000 pounds per square inch) for tooling problems. The thickness of metal which can be handled by plastic forming dies depends to a large extent upon the severity of the forming operation. If the designer has specified broad radii and easily shaped curvatures, considerable thickness may be handled. Sheet materials such as 1/16-inch stainless steel have been successfully handled in a drawing operation with plastic dies.

With a male and female contoured die of nonresilient material, there is a chance for high localized stress concentrations, leading to early failure of the die, unless careful machining brings about a perfect match of die members. Consequently most cast phenolic metal-forming dies are employed in conjunction with a rubber punch in a hydropress.

Cast plastic tools at best are not designed for large

Fig. 4—Below—Installed with its pressure pad, this plastic punch is shown producing top cowl skins on a 750-ton hydraulic press

Photo, courtesy Curtiss-Wright Corp.

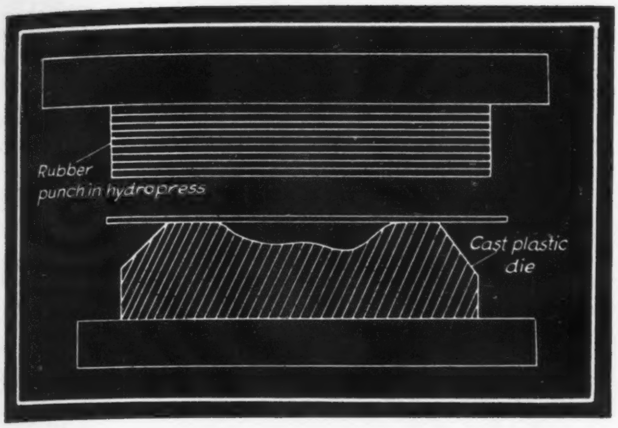
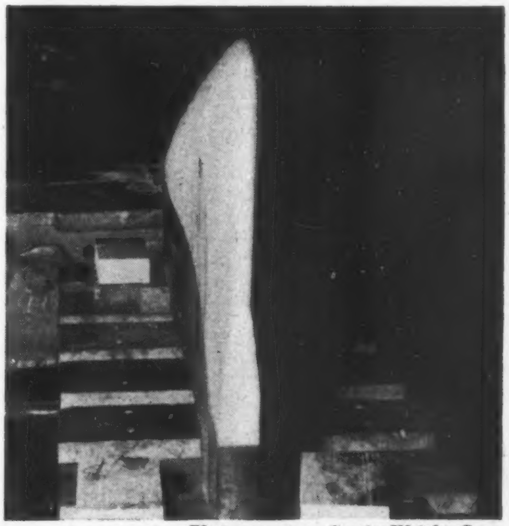


Fig. 5—Above—Rubber punch in conjunction with cast plastic die gives satisfactory reproduction of detail and reasonable die life

production of metal parts, but nevertheless are ideally suited to more limited production involving a few hundred to a few thousand parts. The tougher thermoplastic punches can be expected to turn out many more pieces than the more brittle cast phenolics (impact strength 2 to 3 inch-pound izod notch test). However, production runs up to 1000 or more have been reported on cast phenolics before repairs were necessitated. Sometimes steel or brass tubing is inserted as reinforcement for the plastic casting, giving prolonged life to the unit.

Representative behavior of some of these materials is exhibited in *Fig.* 3 which compares the compressive stress-strain characteristics of an inorganic cement (Formrite), a cast phenolic, and a hot-melt thermoplastic (ethylcellulose). The lower modulus of elasticity of the hot-melt ethylcellulose is reflected in its resilient rubber-like qualities whereas the short deformation of the other materials before failure indicates more brittle qualities. These data will indicate the somewhat limited compression characteristics of cast plastics as compared to metals. The inorganic cement on the other hand, while more brittle, will permit a greater range of temperature in handling metal work.

Cast phenolics, particularly, have hard and well-polished surfaces. These may readily be buffed and polished to any degree of luster. Such application as the forming of transparent thermoplastic sheets into bomber noses benefit greatly from polished surfaces which leave a minimum of mark-off on the formed and drawn sheets.



Photo, courtesy Curtiss-Wright Corp.

Fig. 6—Right—Plastic punch, used to produce wing - tip skins, indicates depth of plastic cast, its thin section and the severe forming operation which it accomplishes

Plastic materials interest machine designers not so much because of their physical properties but because of their *castable* qualities. If the material may be cast against a pattern or model and reproduce all details and dimensions, there are many advantages measured in terms of reduced machining time and costs. This means greater ease of producing experimental samples and less costly procedure if design changes are entailed.

In a recent production run at the Curtiss-Wright plant, one of the fuselage parts for the Commando airplane required four hours production time. The contour was complicated by a reverse curvature and was produced on power presses. In redesigning the tooling for the part, a male and female cast resin die applicable to a stretch press was prepared. Early experimental trials are reported as turning out a perfectly formed piece every fifteen minutes. Further experience has shown that the smooth, slippery cast phenolic sur-

TABLE I

Nonmetallics for Cast Tooling

Material	Cure of Casting Conditions	Tensile Properties	Compression Properties	Impact Properties	Surface Finish	Shrinkage of Unfilled Material	Cost	Status
Phenol-formaldehyde liquid casting resin	Several hours at 170-180 F	Fair	Good	Poor	Good	Slight	High	In production applications
Plaster of Paris	Mix with H_2O and pour at room temperature. Allow to set	Poor	Poor	Poor	Poor	Excellent	Low	In production applications
High-Strength plasters	Mix with exact proportion of H_2O, pour and set at room temperature	Poor	Good	Poor	Poor	Excellent	Low	In production applications
Plaster—Heat-curable, water-soluble resins	Mix with H_2O and catalyst. Pour and set at room temp.	Poor	Good	Poor	Good	Slight	High	Limited production
Plaster of Paris impregnated with Resin X	Impregnate the finished plaster of Paris in Furane Resin X and heat cure	Poor	Good	Poor	Good	Excellent	Low	Experimental
Inorganic cements (Formrite)	Mix liquid and solid phases and allow to set at room temperature	Poor	Good	Poor	Fair	Excellent	Medium	In production applications
Hot-melt thermoplastics—Ethylcellulose	Heat plasticizer and cellulose derivative and pour	Poor	Good	Good	Fair	Slight	High	In production applications
Heat welded thermoplastics	High temperature welding of thermoplastics	Fair	Good	Fair	Good	Excellent	Medium	Experimental

face aids deep-drawing operations. One of the many plastic punches used in this plant is shown in *Fig.* 4.

Cast plastic materials are distinguished from moldable plastic materials in the sense that they may be poured at zero pressure, whereas molding materials require high pressures. Both, however, may require heat to cure them. The mixing and curing conditions of representative cast resins and plasters are compared in TABLE II.

Plastic materials lend themselves to reworking or recasting if design changes are involved. If, for example, the shape or dimensions on a given metal part are to be altered, the corresponding section on the cast plastic tool may be ground or cut out, a suitable pattern inserted and a new section poured into place. If material is thermosetting, additional cure may be necessary.

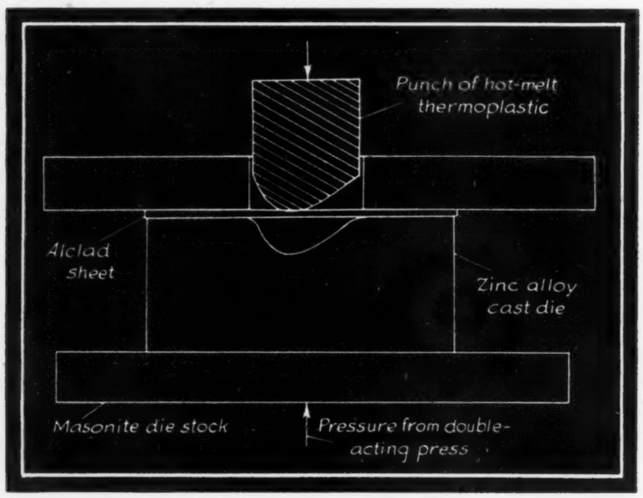
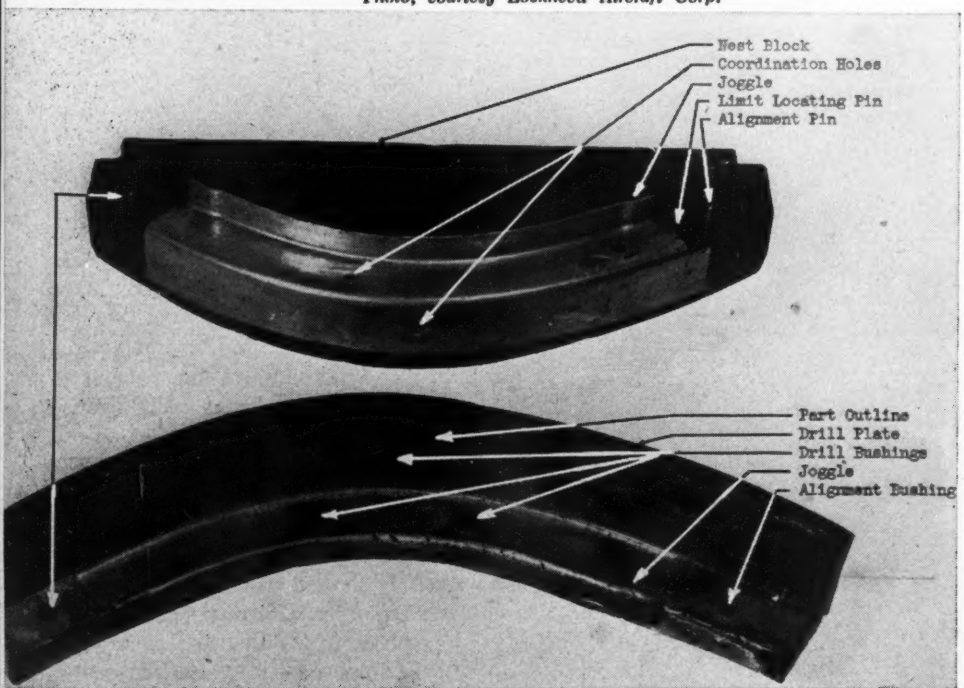


Fig. 7—Above—Hot-melt thermoplastic punch, because of its resilience, is successfully used with metal die

Fig. 8—Below—Drill jigs for parts of irregular shape are among the most popular applications of plastic tooling

Photo, courtesy Lockheed Aircraft Corp.



Reasonably large pieces have been prepared from cast phenolics as well as high-strength plasters. Some phenolic dies weigh over a thousand pounds (specific gravity of material is 1.34). Shrinkage is always a problem and for heat polymerizable materials there is after-shrinkage as well as mold shrinkage. Cast phenolics may have mold shrinkage values of .006 to .007-inch per inch, and service or after-shrinkage of the same magnitude. The latter is due to continued polymerization and some loss of residual water. This problem can be largely coped with through the addition of asbestos fillers, which should reduce shrinkage about 75 per cent. Even the softer thermoplastic tools will undergo some deformation during service, due in part to creep accelerated by the higher temperatures which they reach in forming a metal stamping. To maintain accurate tolerances it may be desirable to scrap the tool every several months, otherwise the designers will have to allow tolerances of the same magnitudes as after-shrinkage changes. Thermoplastics may be successfully remelted and poured.

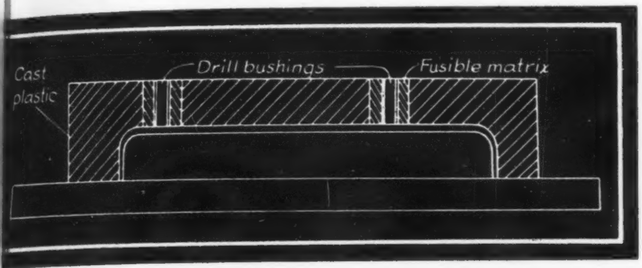
Factors To Be Considered

From a designer's viewpoint, the low cost, availability and ease of manufacture of cast plastic tools will encourage the use of more elaborate tools, particularly for experimental work. The most important factor, which in some cases necessitates more liberal tolerances on sheet metal parts, is the after or service shrinkage of the plastic. Also, the co-efficient of thermal expansion varies from 3 to 9×10^{-5} per degree Cent., depending on the type of filler employed—a factor which may have a bearing in localities where extreme temperature changes occur. Some plastic dies are over ten feet in length, and thermal changes must be considered.

Use of cast plastic tools has not necessitated any major changes in design practice for metal parts. They have been formulated to give exactly what the designer has always specified for metal tools, only at a lower cost and with less difficulty. Applications of cast plastic tooling are many, and if a few of these basic forms are examined, the singular advantages will be apparent. Early experiments in the use of male and female dies of hard cast plastics in hydropress work involving the forming of sheet metal did not prove too practicable, inasmuch as stress concentrations were particularly severe, in some instances leading to cracking of the plastic. However, the reproduction of a cavity detail in cast plastic and the forming of the sheet metal by a thick rubber pad, as in the Guerin process, gave better results. Life of the plastic tooling is extended under these circumstances when the plastic is contained within a steel shell or retaining ring. This example of hydropress forming with a rubber pad is shown in

Fig. 5. The severity of forming that can be accomplished is well illustrated by the draw-die punch shown in *Fig.* 6.

Surface of the plastic can be finished quite smoothly and, if proper care is taken in the preparation of the pattern or model from which the casting is made, little or no further machining is required after the resin is cured. Inorganic cements, high-strength plasters, resin-reinforced



plasters, and cast phenolics have been used in this type of work. Other plastics also have been used successfully in hydropress work. These include lignin types such as Masonite die stock or laminated phenolic plastics. These are furnished in sheet form and are machined or built up to the required size, and used in conjunction with other types of plastic.

Thermoplastic punches for drop-hammer work have made outstanding progress. In an all-metal construction of a drop-hammer die, lead poured to a Kirksite die must be laboriously scraped to allow clearance for the metal thickness. When a castable, hot-melt thermoplastic is substituted for the lead, no allowance is made for metal thickness, the low modulus of elasticity (see *Fig.* 3) permitting sufficient spring-back to adjust to the metal part.

TABLE II

Typical Mixing and Curing of Cast Resins and Plasters

Material	Preparation	Cure
Liquid cast phenol-formaldehyde resin	Add catalyst and filler if desired	Cure at 170-180F, for several hours and cool
Ethylcellulose hot melts	Heat to temperatures around 400F, cool slightly and pour	Cool under slight pressure after pouring
High-strength plasters	Add to required amount of water and pour	Sets rapidly but excess moisture must be evaporated
Dry plaster forms—liquid impregnants	Immerse in impregnant	Cure at 200F. for several hours

Practice has consistently demonstrated that the plastic punches will outlive the lead punches and give cleaner, less noisy operation. In drop-hammer work with plastic punches about two blows are required to form and one good blow to set the piece to final dimensions. Stretch presses have lent themselves admirably toward the formation of metal parts, in conjunction with plastic dies.

As with metal dies, double-acting presses often are used with advantage in successful applications of plastic tooling. A recommended plastic die component for this operation is depicted in *Fig.* 7. Blanking and trimming operations are not performed with plastics directly; though cast plastics may form the body of the tool, the actual shearing edges should be of tool steel.

Among the most popular uses for cast plastics have been their applications to drill jigs, as shown in *Figs.* 8 and 9. Drill bushings are embedded directly in the plastic at the time of casting, and are held in correct contour with the piece for which the drill jig is being prepared. Alternatively, the bushings are located after the casting is shaped and cured, if it is difficult to compensate for mold shrinkage. Because the snug perfect fit of a part to be drilled can be accomplished in no better way than through a cast plastic drill jig, designers may now specify more accurate tolerances on the location of drilled holes on contoured surfaces. Some manufacturers have found it expedient to allow for after-shrinkage by locat-

Fig. 9—Left—Location of drill bushing in a fusible core enables it to be melted out and accurately relocated

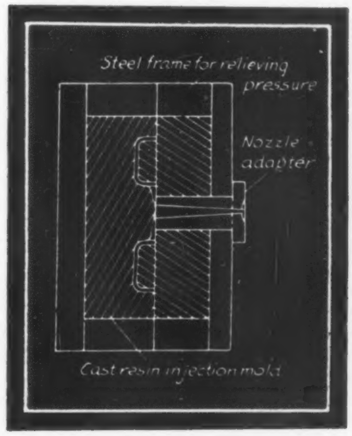


Fig. 10—Right—Low-cost injection mold suitable for experimental moldings is made of cast resin, assembled with metal supporting blocks

ing the drill bushing in a fusible core of Cerromatrix (*Fig.* 9) which can be melted out and new tolerances established after a long lapse of time. Drill jigs also may be prepared from thermoplastic scrap available in most aircraft plants. This involves high-temperature fusion of ground scrap such as may be obtained from windshields, bomber noses and the like. Machine designers will no doubt specify an increasing number of plastic drill jigs and contour blocks to be employed in checking production work on metal parts.

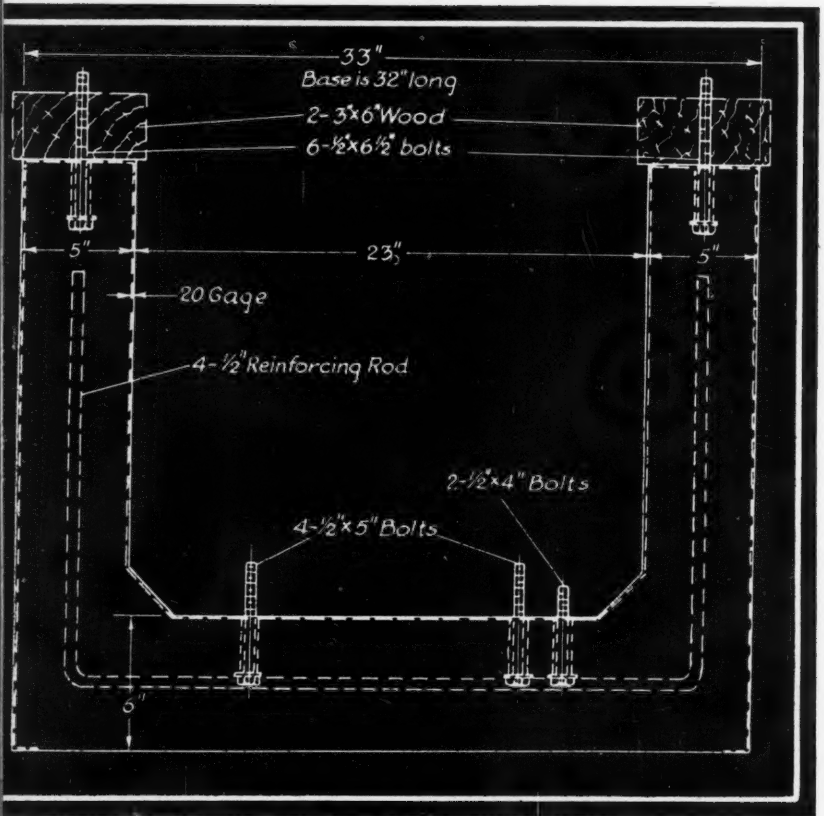
Cast Plastics Found Superior

Other holding fixtures and form blocks which have found cast plastic materials to be far superior to any machined form employed in the past include plastic castings for molding irregularly shaped forms to be mounted in a lathe or chuck, and cast forms for routing jigs. One further application of note is the preparation of cast resin injection molds. Their assemblage with metal supporting blocks, shown in *Fig.* 10, makes available a low-cost injection mold suitable for experimental moldings.

Sometimes metal protection has been added to the surface of the plastic either by a special electroplating process or by metal spraying, though the added time and labor often counteracts the initial benefits.

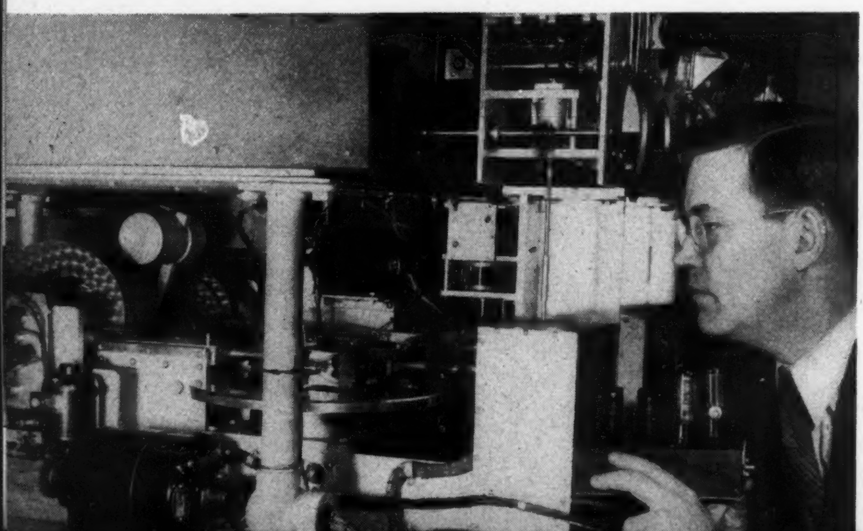
While increased production and lower tooling costs have been the highlights of the cast plastic tooling program, the future importance of these developments encompasses a much greater field. As already mentioned, designers of machine parts faced with the production of experimental models of their design encounter tooling costs so high in many cases as to necessitate hand samples. Availability of low cost, serviceable plastic tooling will permit earlier evaluation of their ideas.

Scanning the field for IDEAS



Concrete-filled machine base proves highly economical, conserves cast iron and effectively absorbs vibration. The base shown at left for a thread milling machine, designed by The Fred Goat Co., Inc., was produced for the nominal cost of $12 for the shell and $5 for concrete. Compared with a cast base, this effected savings for casting, machining and making a pattern. The shell serves as a form for the concrete and is painted in the conventional manner, appearance of steel-fabricated construction being achieved.

Formed on a brake the U-shape shell is 20-gage sheet, open at each end to provide space for a motor and other parts. Mounting bolts are positioned through drilled holes in the steel sheet and utilize spacers of standard piping. The concrete, a 1-1½-3 mixture, is poured into the base in inverted position. Reinforcing rods are ½-inch rounds, placed in position during the pouring operation.

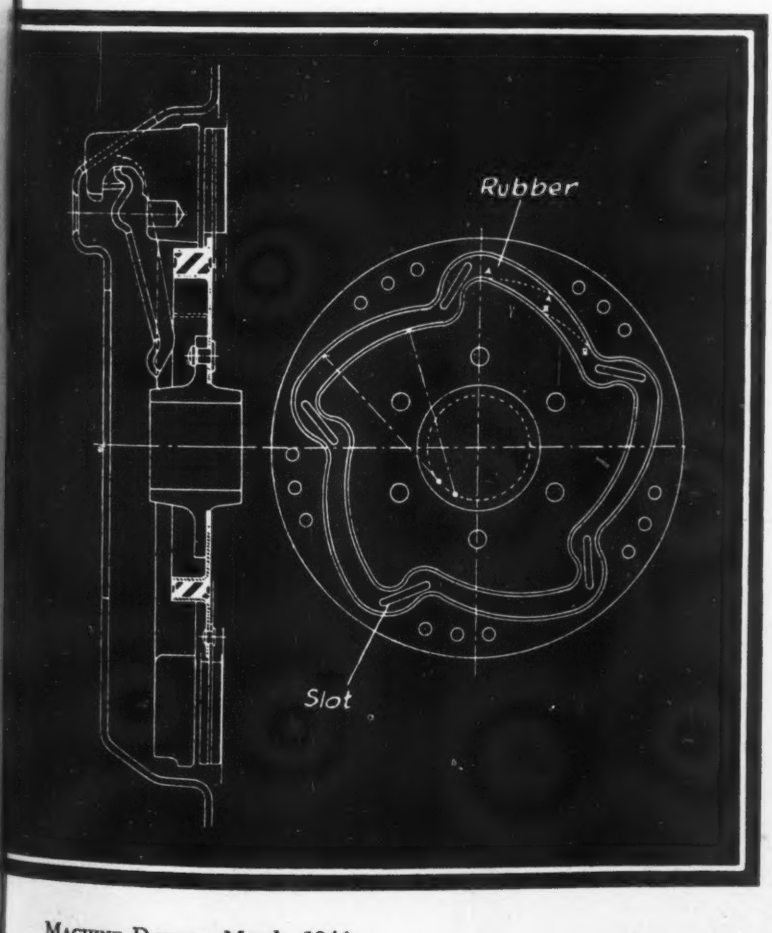


Calculating machine, left, tests samples of turbine nozzles by measuring, recording and adding up energies at various points on the nozzle. Compressed air instead of steam is blown through the nozzles. In this mechanical brain, designed by General Electric, the varying pressures move tiny upright wheels back and forth on a disk as it turns. The rotation of these wheels is conveyed by rotating shafts of light,

through a light polarizer, to photoelectric tubes. These tubes control a series of cams and gears in the form of a servo-mechanism which operates a printer. The printer in turn adds the energies and indicates overall energy loss.

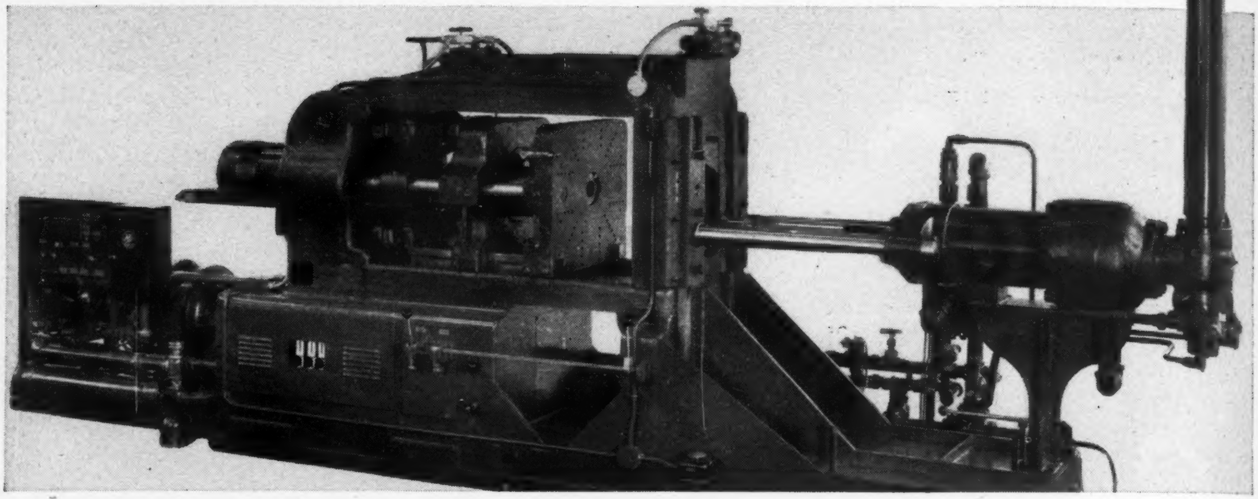
High-speed motor, right, obviates use of step up devices to obtain 60.000 revolutions per minute for grinding, polishing and routing machines. The motor, designed by Westinghouse, is an induction type and is supplied with power from a 1000-cycle generator. Centrifugal stresses at the surface of the rotor, which is $1\frac{3}{4}$ inches in diameter and 2 inches long, reach about 15,000 pounds per square inch when turning at this speed.

Elastic driving element for incorporation in a clutch disk utilizes molded rubber as shown below. Geometry of the molding is selected to fit each particular drive both for the driving condition and for the condition under which the clutch tends to overrun. In this coupling, designed by Metalastik, Ltd. of Leicester, England, and reported in "Engineers' Digest" the characteristics of rubber are particularly advantageous for both conditions of operation. High yield, high resilience, and viscosity of internal friction are particularly valuable for the damping characteristics needed.



The rubber element is subject to both shear and compression, but as the torque increases more resistance is taken up in compression. Rubber fills the space between the cooperating curved surfaces and is bonded to each. To provide uniformity of stress, both inner and outer curves have the same radius, struck from a base circle as indicated in the drawing. Slots are formed in the rubber at the sections of small curvatures to distribute the stress when the clutch tends to overrun.

High pressure for hydraulic systems requiring a squeeze at the end of the power stroke is effectively obtained in the die-casting machine shown at top of next page. Designed by Lester Engineering Co. for Lester-Phoenix, the system consists of a hydraulic cylinder fitted with a large actuating piston, having a hollow piston rod which contains a passageway to a smaller inner fixed piston. Oil under 1000 pounds per square inch pressure is introduced through the hollow rod, displacing at high velocity the small piston opposing it. This



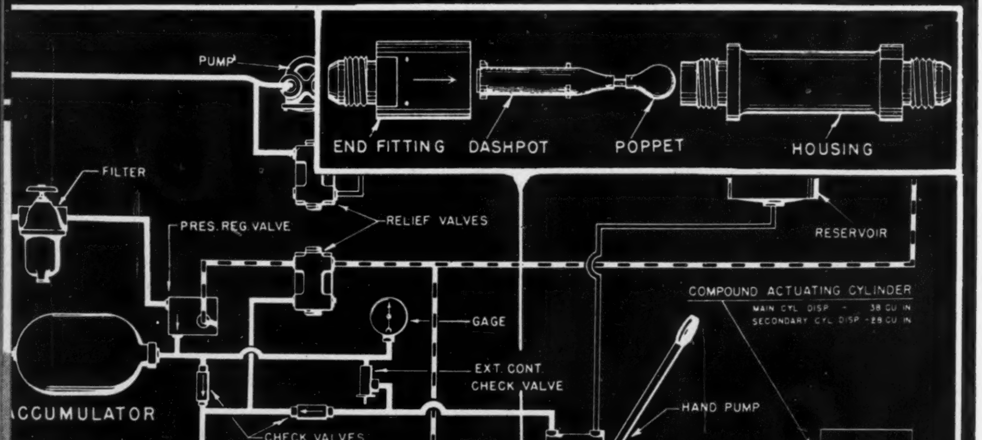
piston carries with it the piston rod and the attached main or large piston.

As the latter moves forward at high speed, oil flows by gravity from a vertical storage tank through a check valve to occupy the space back of the large piston. When the die cavities have been filled, 2000 pounds pressure from a booster pump is applied direct to both pistons resulting in a high sustained final pressure which packs the metal into the die, producing a strong dense casting.

Protection of hydraulic lines from undue loss of fluid through ruptured piping, such as encountered in aircraft during combat, is assured through the use of quantity-metering devices installed in the lines to each cylinder. Called "hydraulic fuses", these units are applied as shown below. Each is calibrated to measure a definite quantity of oil so that if the amount passed exceeds the capacity of the cylinder served, the device automatically shuts off the flow.

As fluid under pressure flows in the direction shown in the disassembled view, it pulls with it a poppet. Resistance to this poppet motion is effected by a check-valve dashpot. Dashpot oil is allowed to escape through a small orifice, the size of which determines the poppet motion or the quantity of oil allowed to pass before the poppet is seated. Reversal of flow resets the fuse by reversing the action. The poppet, however, will reset more readily because of a resetting spring and the opening of the ball check. Designed by Simmonds Aerocessories, Inc., the fuse is little affected by variations in temperature, viscosity or rate of flow. Back pressures, surges, vibration and mounting position also have little effect and are within the operating limits of the unit.

For maximum protection the fuses are placed as close to the valves as possible, as shown in the sketch. If one valve operates more than one cylinder, a fuse for the total capacity is installed at the valve with the added protection of cylinder-capacity fuses at the junction of each branch. Regardless of where a break may occur, maximum protection is then assured for the undamaged parts of the system.



Frequently Overlooked Factors in Design of Springs

By A. M. Wahl
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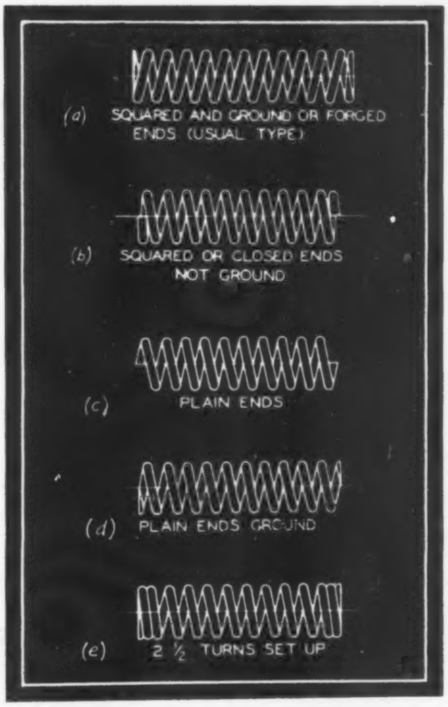


Fig. 1—Typical forms of ends for helical compression springs

***BASED ON** a chapter in the author's forthcoming book on mechanical springs, this article discusses some of the pertinent factors involved in design of helical compression springs. These factors, which frequently are overlooked, include allowance for end turns, effect of eccentricity of loading, variations in modulus of rigidity, effects of cold setting, and stress at solid compression*

PERHAPS the most important consideration involved in the design of a helical compression spring for a given application is the proper choice of working stress. However, there are a great many other factors which frequently are overlooked but which are, nevertheless, important. Such factors include design of end coils, allowance for end coils, effects of eccentricity of loading, variations in modulus of rigidity, effects of cold-setting operations, design stress at solid compression, and others. It is the purpose of the present article to discuss the effects of some of these factors in the light of the results of recent research.

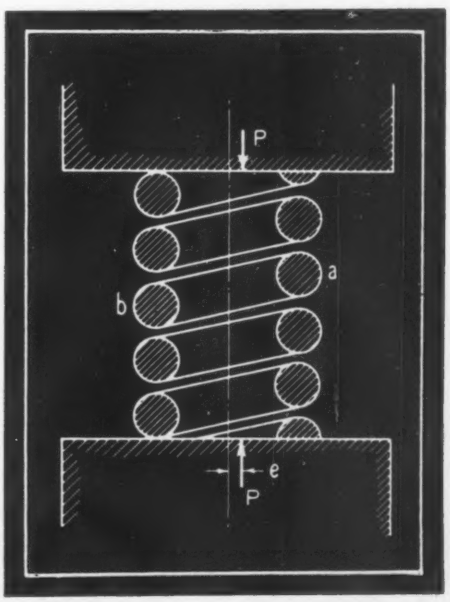
Some of the more usual types of end turns are indicated in *Fig.* 1. The most common of these have ends set up and are ground (or forged) as indicated in *Fig.* 1*a*. This type of end turn has the advantage that there is less eccentricity of loading (and hence a lower stress for a given load) than would be the case where the ends are made as indicated in *Fig.* 1*b*, *c* or *d*. In *Fig.* 1*b* the ends are simply squared and closed, while in *Fig.* 1*c* they are left plain without any grinding. This spring would give the highest amount of eccentricity of loading. The spring of *Fig.* 1*d* is the same as that at *c* except that the ends have been ground so that at least ½-turn at each end is flat. In *Fig.* 1*e*, a spring with 2½ turns set up and ground is shown.

Determining Effect of End Turns

An accurate determination of deflection in helical compression springs requires that the effect of the end turns be estimated with reasonable accuracy. Some experimental and analytical work by Vogt(1)* indicates that for the usual design of end coil with ends set up and ground (*Fig.* 1*a*) the number of active coils is equal to the number of completely free coils plus ½. Thus if a compression spring has 10 free coils and 12 total coils (tip to tip of bar) then on this basis the number of active coils would be 10½, and ¾ of a turn would be inactive at each end. However, when the load is increased there is some progressive seating of the end turns so that the number of completely free coils decreases with the load, and this increases the number of inactive turns.

Pletta, Smith, and Harrison(2) made a series of careful tests on com-

Fig. 2—Eccentricity of loading which results when compressing a helical spring between parallel plates. Stress at (b) is greater than at (a) due to eccentric load



*References in parentheses are listed at end of article.

mercial springs in which they used a special setup to determine end-turn effect. The results of these tests indicate that *at zero load* the number of active turns was equal to $n' + \frac{1}{2}$ where n' is the number of completely free coils at zero load. However, as the load increases the number of inactive coils was found to increase due to seating of the end coils, the amount of increase varying from .5 to 1 turn at usual working loads, with an average of about .7-turn. Since for calculation purposes the average number of active turns in the range from no load to working load is of primary interest (this would be used in the deflection formula) it appears reasonable to subtract about half of this decrease from the number of active turns. This gives a figure for average active turns varying from n' to $n' + \frac{1}{4}$. Since for the usual type of end turn (*Fig.* 1*a*) the number of total turns is $n' + 2$, this means that the inactive turns found in these tests varied from about 1¾ to 2 with an average of perhaps 1.85.

An analysis made by H. C. Keysor(3) indicates that the total number of inactive coils in the spring is approximately equal to 1.2 as a deduction from "solid turns". This is based on the common practice of taking the number of solid turns equal to the solid height divided by bar or wire diameter. Since for the usual shape of end coil, the solid turns are equal to the "total turns" measured from tip to tip of bar, minus ½-turn, this figure of 1.2 would be equivalent to a deduction of 1.7 turn from total turns.

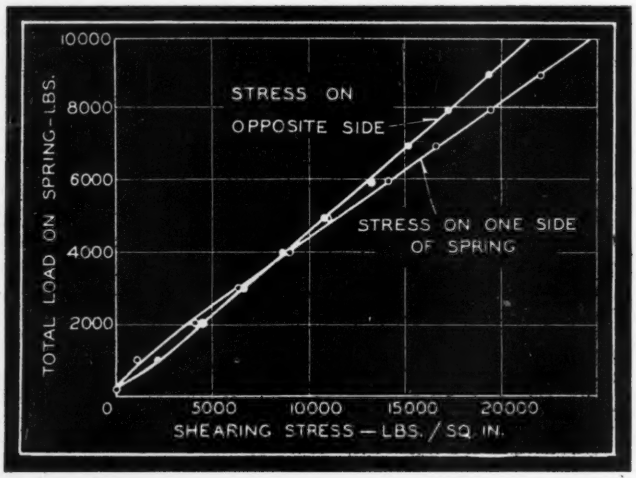


Fig. 3—Load stress curves on spring tested as in Fig. 2

Some additional data on inactive turns were given by Edgerton(4) based on the research of the special research committee for mechanical springs of the A.S.M.E. The average value obtained by Edgerton was 1.15 as a deduction from solid turns or 1.65 as a deduction from total turns.

Taking the results of these various investigations as a basis it appears that, depending on the load, the number of inactive coils may vary from about 1.65 to 2 considered as a deduction from total turns. Probably a mean value of 1¾ inactive coils, considered as a deduction from total turns would be as good a figure as any to use in practice. For higher loads, possibly a figure somewhat larger may be justified, while a lower figure may be used for smaller loads. The seating of the coils as the load increases also tends to produce a slight curvature of the load-deflection diagram. Fur further details, the reader is referred to the references mentioned and, in particular, to the investigation by Pletta and his associates(2).

The preceding discussion has been concerned only with the usual type of end turn, *Fig.* 1*a*. Test results concerning the other type of end coils shown in *Fig.* 1 are lacking, but approximate values of inactive coils may be used as follows: For plain ends (*Fig.* 1*c*), active turns are $n - \frac{1}{2}$ where n = total turns; for plain ends ground, *Fig.* 1*d*, active turns are $n - 1$. If 2½ turns at each end are set up and ground, as in *Fig.* 1*e*, the active turns may be taken roughly as $n - 5$. These values should be considered very rough.

Eccentricity of Loading

One effect which is frequently overlooked by spring designers is that of eccentricity of loading in actual springs. Thus if a compression spring with the usual type of end coils, *Fig.* 1*a*, is compressed between two parallel plates as in a testing machine, it will be found that in general the resultant load P is displaced from the spring axis by a small amount e as indicated in *Fig.* 2. The effect of this eccentric loading is to increase the stress on one side of the spring diameter and decrease it on the other. This condition is clearly indicated by the load-stress curves of *Fig.* 3 which were obtained by putting extensometers on an actual helical spring of 1½-inch diameter bar. The full circles represent stress on one side of the coil while the open circles that on a diametrically opposite side. It may be seen that due to eccentricity of loading, at the higher loads the stress on one side is considerably greater than that on the other.

TABLE I

Tests To Determine Eccentricity of Loading in Helical Springs

Spring No.	Spring O.D. (in.)	Wire diam. (in.)	Turns between tip contacts (n')	Total load (lb.)	Ratio e/r — Test	Ratio e/r — Calculated*
0	2⅞	.177	4	34	.12	.12
1	2⅞	.177	4	34	.04	.12
2	2⅞	.177	4¼	31	.09	.11
3	2⅞	.177	4½	29	.14	.11
4	2¾	.177	2	38	.19	.23
5	2¾	.177	2	38	.13	.23
6	2¾	.177	2	38	.24	.23
7	2¾	.177	2¼	32	.22	.21
8	2¾	.177	2¼	32	.27	.21
9	2¾	.177	2¼	29	.13	.21
10	2¾	.177	2½	30	.30	.19
11	2¾	.177	2½	30	.22	.19
12	2¾	.177	2½	30	.20	.19

*Using equations for Z and e given in Keysor's paper and taking $N = n' + 1.5$.

An analysis of the effect of this eccentricity of loading based on certain assumptions has been carried out by Keysor(3). Because of the complexity of the analysis, it will not be given here. However, the final results are summarized in the curve of *Fig.* 4, the ordinates representing ratios e/r between eccentricity of loading and mean coil radius and the abscissas number of turns n' between tip contact points. (The total turns n for the usual design will be equal to $n' + 2$).

It is seen that the eccentricity ratio e/r fluctuates be-

tween zero and maximum values, the zero values occurring approximately at $n'=\frac{1}{2}, 1\frac{1}{2}, 2\frac{1}{2}$,—etc. Theoretically it should be possible to get pure axial loading (i.e., zero eccentricity) by choosing n' to conform with these values. However, because of variations in actual springs and possibly also because of inaccuracies in the assumptions made in the analysis, axial loading cannot in general be realized in practice(5). For practical design, therefore, the envelope of the curve as indicated in *Fig.* 4 should be used.

For calculating the ratio e/r the following expressions given by Keysor(3) may be used:

$$e/r=1.123(Z-1) \quad \text{(1)}$$

$$Z=1+\frac{.5043}{N}+\frac{.1213}{N^2}+\frac{2.058}{N^3} \quad \text{(2)}$$

In these, $Z=$ eccentricity stress factor, $N=$ number of solid coils. This will be approximately $1\frac{1}{2}$ turns greater than the number of coils n' between tip contact points, i.e., $N=n'+1.5$. By using these equations the ratio e/r may be calculated for any given N. As an approximation it may be assumed that where the spring index is fairly large the stress will be increased in the ratio $1+e/r$ as compared with the stress for purely axial loading.

Some tests were made several years ago by the writer which give a rough check on this formula. These were carried out in connection with an application where it was desired to obtain as nearly as possible a central load on a helical compression spring. The tests were made on small helical springs using a special three-point loading fixture so arranged that the eccentricity of loading could be determined. Essentially this consisted of a flat plate with provision for attaching dead weights 120 degrees apart on equal radii. When equal loads were applied at equal radii, it was found in general, that the loading planes at each end of the spring were not parallel. The loads were then adjusted to give parallelism of these loading planes; from the magnitude of the required loads the eccentricity of loading could be calculated.

The results of these tests are summarized in Table I. The outside diameter, wire diameter, number of turns n'

Table II

Allowable Variations in Commercial Spring Wire Sizes

Type of Material	ASTM Specification (No.)	Wire Diameter (in.)	Permissible Variation (in.)
Hard-Drawn Spring Wire	A-227-39T	.028 to .072	± .001
Chrome-Vanadium Spring Wire	A-229-39T	.073 to .375	± .002
Oil-Tempered Wire	A-231-39T	.376 and over	± .003
Music Wire	A-228-39T	.026 and under	± .0003
		.027 to .063	± .0005
		.064 and over	± .001
Carbon-Steel Valve Spring Wire	A-230-39T	.093 to .148	± .001
Chrome-Vanadium Valve Spring Wire	A-232-39T	.149 to .177	± .0015
		.178 to .250	± .002

between tip contact points, and load also are given. The springs tested had ground end coils of the usual form so that the total number of turns equaled $n'+2$. In the last column the values of the ratio e/r between eccentricity and coil radius as calculated from Equation 1 are given. For comparison the test values of e/r as measured on these various springs are also listed in the next to the last column.

It will be seen that in most cases the agreement between calculated and average test values is sufficiently good for practical use, especially if it is considered that the test springs were handmade and no particular care was taken in forming the end turns.

For best results the increase of the stress due to eccentricity of loading should be taken into account in design, particularly if the number of coils is small.

Variations in Spring Dimensions

Another factor often overlooked by the designer is the effect of slight variations in spring wire or coil diameters on the load-deflection characteristic of helical compression springs. Since the deflection varies as the third power of the mean coil diameter, a 1 per cent change in the latter means a 3 per cent change in the load deflection characteristic. Similarly a 1 per cent change in wire or bar diameter will result in a 4 per cent change in the load-deflection characteristic. This means that small

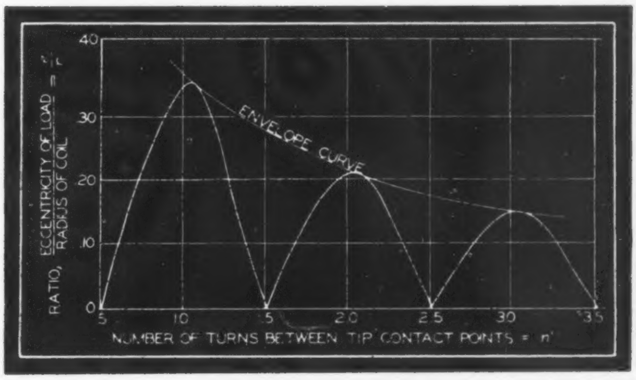


Fig. 4—Ratio of eccentricity of loading (e) to coil radius (r). Total number of turns for usual design is (n'+2), number of solid coils is (n'+1½)

changes in these dimensions will result in relatively large changes in the spring constant.

To give an idea of the possible variations in wire diameters to be expected in practice, Table II shows allowable variations in commercial wire sizes obtained from data in the A.S.T.M. Standards.

As may be seen from this table, music wire of .064-inch diameter may vary by ± .001-inch from the nominal value or by about 1½ per cent. Such a variation means a 6 per cent possible change in the load-deflection characteristic. However, if the wire is 1½ per cent undersize the spring maker might compensate for it by reducing the coil diameter by 2 per cent or by a slight reduction in number of turns. In this manner the spring load-deflection characteristic may be brought back to the design value. If springs are to be held to relatively close limits as far as deflections are concerned it is well, if possible, to allow the spring maker some leeway in the choice of the coil diameter or on the total number of turns. By this means, compensation for variations in wire size may usually be obtained without sacrificing es-

sential spring characteristics and without excessive cost.

In winding springs cold, there is always some "spring back". In other words, the inside diameter of the springs after winding will be slightly greater than the diameter of the mandrel as a consequence of the elastic properties of the material. Although this effect theoretically may be compensated for by using a slightly smaller mandrel, in practice variations in the elastic and plastic properties, even in the same kind of material, will result in variations in the coil diameter. As an example of the amount of such variation to be expected, the tolerances given by one manufacturer (6) for close cold-wound springs are listed in Table III. It may be seen that these depend on the spring index D/d and on the mean coil diameter D.

Table III

Tolerances on Spring Coil Diameters*

(Close Cold-Wound Helical Springs)

Mean Coil Diameter, D (in.)	Variations in Diameter D: $D/d = 4$ (± in.)	$D/d = 8$ (± in.)	$D/d = 12$ (± in.)
1/8	.003	.0035	.005
1/8 to 3/16	.0035	.005	.0065
3/16 to 1/4	.005	.0065	.0085
1/4 to 3/8	.0065	.0085	.0105
3/8 to 1/2	.0085	.0105	.0130
1/2 to 3/4	.0105	.0130	.0155
3/4 to 1	.0130	.0155	.0230
1 to 1 1/2	.0155	.0205	.0318
1 1/2 to 2	.0185	.0313	.0408
2 to 3	.030	.0430	.0528
3 to 4	....	.0550	.0730
4 to 5	....	.0725	.095
5 to 6	....	....	.125
6 to 7	....	....	.165
7 to 8	....	....	.210

*Data from American Steel and Wire Co. *Manual of Spring Engineering.*

Examination of this table shows that variations in coil diameter as much as 5 per cent may be expected for the larger indexes $D/d = 12$, and as much as 3 per cent for the smaller indexes $D/d = 4$. Such variations may result in a 9 to 15 per cent change in spring constant unless compensated for by changes in number of turns or wire size.

Effect of Modulus of Rigidity

One of the factors which the designer should keep in mind, particularly if an accurate load-deflection characteristic is wanted, is the effect of variations in the modulus of rigidity or torsional modulus for different materials.

Table IV

Average Values for Modulus of Rigidity

Spring Materials

Material	Modulus of Rigidity G (psi)
Music Wire; Carbon- and Chrome-Vanadium Steels	11.5×10^6
Hard-Drawn Stainless Steel	10.5×10^6
Monel Metal	9×10^6
Phosphor Bronze	6×10^6

This problem has been discussed more fully in a previous paper(7), a survey of the literature showing that the modulus values may vary by several per cent for carbon spring steels. In particular, the effect of a decarburized surface layer only a few mils thick is such as to reduce the effective torsional modulus by a considerable amount. On the basis of available test data a good average figure for modulus of rigidity for carbon and alloy steels is 11.5×10^6 pounds per square inch. However, for hot-wound carbon steel springs in the larger sizes some manufacturers recommend modulus figures as low as 10.5×10^6 pounds per square inch. Presumably this is done to compensate for the effect of surface decarburization on the basis of available test data. The modulus of rigidity values given in Table IV may be used for various spring materials. However, it should be noted that in individual cases deviations of several per cent may occur from these values.

Stress at Solid Compression

In the design of compression springs it is desirable to choose the coil pitch such that when the spring is compressed solid no appreciable permanent set will occur. The reason for this is that usually in operation the spring may at times be compressed solid and if, under these conditions, it takes a set the load at the working deflections will be changed. Thus the spring will no longer have its initial characteristics. For most machines and devices, such permanent set is objectionable.

If a compression spring is initially wound with a coil pitch sufficiently great so that the elastic limit of the material is exceeded when the spring is compressed solid, the distribution of stress along a diameter of the cross section is somewhat as shown in *Fig. 5b* for a spring of large index (at low loads before the elastic limit is reached the distribution is approximately linear as shown in *Fig. 5a*). After the load is released the residual stress distribution will be somewhat as shown by the shaded area in *Fig. 5c*. For a spring of large index these residual stresses may be calculated approximately from the condition that the moment of the stress represented by the triangle *obc* must be equal to the moment of the stresses represented by the area *oadc*. The residual stress represented by the vertical line *bd* is in an opposite direction to the load stress. Consequently when normal load is again applied to the spring the peak stress at the surface of the bar or wire is reduced by an amount represented

Table V

Suggested Stresses* at Solid Compression

(Helical Springs)

Material (in. dia.)	Stress at solid compression up to which it is not necessary to remove set (psi)	Maximum stress at solid compression with all set removed (psi)
Music Wire		
up to .032	130.000	180 000
.032 to .062	110.000	170 000
.062 to .125	100 000	160 000
.125 and over	90,000	150,000
Hard-Drawn Spring Wire		
up to .032	120.000	170 000
.032 to .062	100 000	160 000
.062 to .125	90 000	150 000
.125 and over	80,000	140,000
Oil-Tempered Wire		
.125 and over	80,000	140.000
18-8 Stainless (Hard Drawn)		
up to .125	85 000	140.000
over .125	75,000	120,000
Phosphor Bronze		
general sizes	40,000	70,000

*Curvature correction included. (From *The Mainspring* Feb. 1941).

by the line *bd*. The resultant stress distribution at the working load is shown in *Fig. 5d*, where it is seen that a more favorable stress distribution, with lower peak stress, occurs. Consequently the static strength of the spring and also the fatigue life are improved.

In the process of cold setting, a reduction of the initial free length occurs. Thus, to obtain a given free length, the initial value after coiling must be made enough greater than the desired value to compensate for this reduction. If it is desired to use the spring without cold setting, the stresses at solid compression must be kept considerably lower than would be the case if cold setting were used, TABLE V. If this is not done the free length

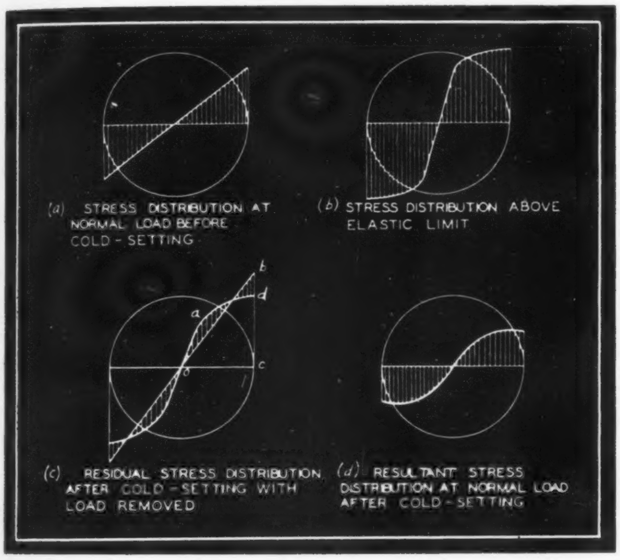


Fig. 5—Effect of cold-setting operations, assuming a large index for the spring

will change after the first solid compression when the spring is in service.

It will be found that beyond a certain limit there will be no additional gain by using the cold-setting process. In other words, beyond a certain initial free length, the final length after the setting operation will be the same. The reason for this is that the stress-strain curve tends to flatten out so that a higher strain does not give an appreciably greater torsion moment. If this point is greatly exceeded excessive cold work and loss of ductility may occur.

"Recovery" Should Be Considered

Another effect which occurs after cold setting is what is known as "recovery". Thus immediately after the settage operation the free length of the spring will be a certain value; however, if on standing for some time the settage stress is too high the free length will increase slightly. The reason for this is bound up with relaxation and creep effects due to the residual stresses present. Change in free length with time is objectionable in most applications.

Suggested stresses at solid compression for various spring materials as given by Wallace Barnes Co.(8) are listed in TABLE V (the values include a curvature correction factor). In the second column the stress at solid compression up to which it is not necessary to remove set is given, while in the last column is given the stress at solid compression which may be used after all set is removed by means of repeated compressions of the spring.

Although the stresses at solid compression with set removed as given in this table may seem extremely high, it should be remembered that they are *calculated* values used mainly for design purposes and which neglect yielding of the material. Hence the actual stresses will not reach these values. If stress concentration effects due to curvature are neglected and if account is taken of the stress distribution as shown in *Fig. 5b*, it will be found that the actual stresses are of the order of magnitude of the yield point in torsion. Consequently calculated stresses such as those of TABLE V may be entirely feasible when determining maximum coil spacing in helical compression springs. This does not mean that *working* stresses equal to these values may be employed. Actually much lower values for working stress should be used depending on the type of spring application. A discussion of these is, however, beyond the scope of the present article.

Careful consideration of the foregoing often-overlooked factors involved in helical compression springs should result in better and more satisfactory designs.

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WPB Issues New Rulings

ELECTRIC motor controller manufacturers have been relieved by War Production Board from the technicality of obtaining an AA-5, or higher, priority before starting production on these units or parts for them, through an amendment to Limitation Order L-250. This permits producers to make economical runs of components for the controllers. The requirement, however, of obtaining an AA-5, or higher priority before acceptance or delivery of completed electric motor controllers remains in force.

♦

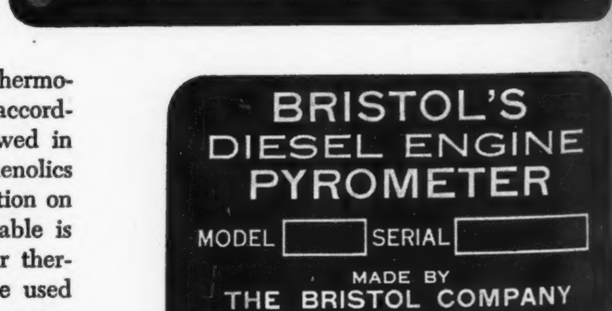
From War Production Board also comes an announcement that users of antifriction bearings may not accept deliveries in excess of specified amounts during any one month of the second and third calendar quarters of 1944 without specific approval. This was done in order to relate the actual needs for such bearings to the supply—and to assure that statements of delivery requirements are realistic.

Selecting Plastic Nameplates

By John W. Greve

Fig. 1—Electrical instrument has its instruction plate printed on the inside of the hinged cover. On the reverse side, the nameplate is molded with the cover in relief

Fig. 2—Below—Etched plate made from laminated plastics. Black face and green core give a two-color plate



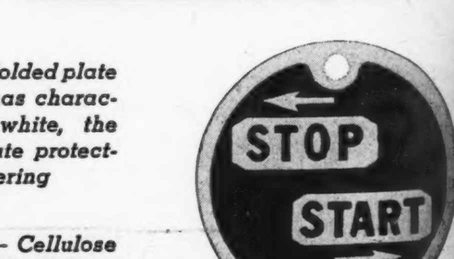


Fig. 3—Above—Molded plate of thermoplastic has characters filled with white, the surface of the plate protecting the lettering

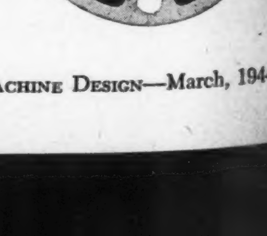


Fig. 4 — Right — Cellulose acetate plate of etched sheet, filled in with black

PLASTICS are successfully replacing critical metals such as brass, aluminum and stainless for nameplates, instruction plates, data plates and dials on many types of machines. Limitations placed on the use of metals for these purposes have been responsible for the application of alternative materials such as plastics, many of which have been applied with gratifying results.

Distinction usually is made in limitation orders between nameplates on one hand and utility plates such as instruction and data plates on the other. No clearly defined rules have been set but it generally is agreed that a nameplate refers to a trademark type of plate, the use of which is solely decorative or of an advertising nature. Utility plates, however, may contain the name and address of the manufacturer as essential information. A rating plate on a motor, for instance, also gives the manufacturer's name, this being given, too, on serial plates.

Such restrictions as apply to plastics concern chiefly the thermoplastic types. These cannot be used for nameplates as such, according to order M-154. Use for utility plates, however, is allowed in this limitation order. Decorative nameplates may utilize phenolics such as the aldehyde types. This use is covered under the section on specialty manufacture in order M-246 and the quantity available is governed by the supplier's quota. The paraphenolics or clear thermosetting plastics are covered in order M-254 and cannot be used because of the high restrictions placed on them for military purposes.

It is generally understood in all of the conservation orders that the most available material that will successfully serve the purpose should be used. Under very adverse conditions this would mean that aluminum or stainless has to be used while in a great many other cases plastic materials of various types would be indicated.

Decorative nameplates should receive a great deal of care in their design because of their advertising and prestige character. Trademarks or emblems can be reproduced accurately on these plates which often have a prominent place on the machine. Util-

Fig. 5—Above—Injection-molded index plate of transparent plastic. Figures are depression molded on back side and are filled in with white paint

Fig. 6—Below—Embossed plates of laminated phenolic, punched and sheared. Enamel filling supplies contrasting color

ity plates, however, are designed principally to be clearly readable throughout the life of the machine. The trend toward modern design permits great possibilities in the choice of contrasting plastic materials that will harmonize with other plastic fittings on the equipment, offering full use of the attractive colors available in plastics.

Representative groups of both thermoplastic and thermosetting materials are in use. Choice of available materials is large but careful study of requirements and proper materials is necessary. Indicative of the possibilities is the instruction plate shown in *Fig.* 1 which is an integral part of the hinged cover. Plates from etched or printed thermoplastics such as cellulose acetate or vinyl sheets, as well as plates using certain plasticized thermoplastics such as urea melamine resins, are readily blanked, punched or sheared. Existing fabricating tools of the metal nameplate trade usually can be employed.

Most common processes used for plastic plates include etching, embossing, molding, engraving and printing. Characters or designs thus formed—with the exception of those that are printed—usually are filled in with colors for desired effects. Special processes are also available such as double-shot molding, heated inking dies, etc. A recently developed method for etching produces economical and attractive plates suitable for a variety of applications, *Fig.* 2. Laminated sheet, having the top lamination a color contrasting from that of the core, is etched to give either an engraved or an embossed effect. Plates utilizing both may be made in a variety of attractive designs. Although requiring specially laminated stock, this method has many advantages and obviates the necessity of filling in with color. By carefully controlling the etching process and by using stock with several colors it is possible to achieve multicolor effects through etching to the depth of the color desired.

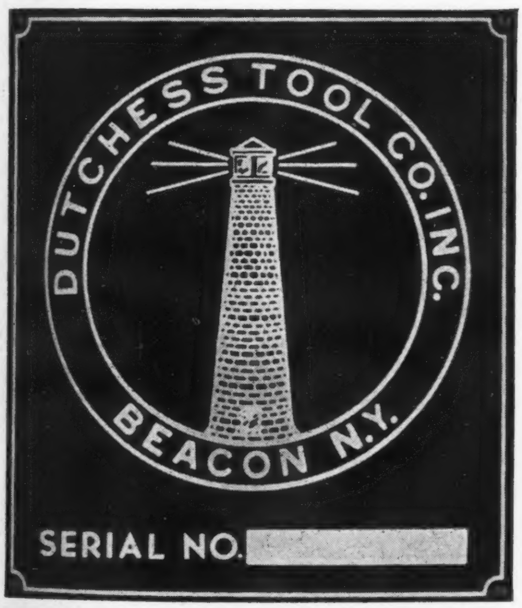
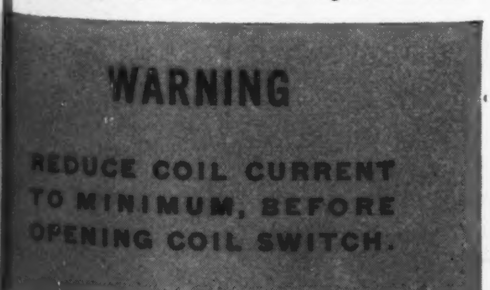


Fig. 7—Above—Engraved plate. Black and white core laminations obviate filling-in operation

Fig. 8—Below—Graphic type of plate utilizing a printed sheet for top lamination



Molded Plates Are Inexpensive

Molded plates, especially of thermoplastics, *Fig.* 3, are utilized widely and may be produced economically if the quantities are sufficiently large. If the characters are in relief no further operation is required. When depressed, however, they may be filled with bright enamel, gold or other coloring material.

Many machines have nameplates cast into the frame or a member instead of employing a separate plate. Draft, of course, is applied to the sides of the characters to provide proper ejection from the mold. Often operating data such as indexes on knobs are cast in and filled similar to that for plates.

Clear or translucent plastics make excellent decorative plates, instruction plates and dials where modern back lighting is employed. Signal lighting for dials, etc., indicating energized circuits is helpful to operators and also improves visibility of the printing. Such plates usually have characters depressed from the reverse side and color filled as in *Fig.* 5.

Embossing may be utilized if the details are not intricate and the amount of relief is small. A typical embossed plate is shown in *Fig.* 6. Engraving, although relatively expensive, is employed for plates requiring accurate detail and for limited quantity production. The characters have clean sharp edges which are ideal for filling in with color;

clearer details, *Fig.* 7, are obtained by this method than by etching, embossing, molding, etc., although the detail obtained by the other methods is considered satisfactory.

Printed phenolic or urea plastic plates such as Formica Permaprint, GE Graphic Laminated and Graphic Lamicoid are produced by using a printed sheet, covered by a film of clear resin, as the top lamination. Silk-screen and photographic processes are also employed, similarly. Resin flows over the surface during the curing process to form a protective coating, *Figs.* 8 and 9. Excellent for wiring diagrams, schematic circuits, etc., such plates will withstand heat up to 275 degrees Fahr. and, having a smooth surface, they are easily wiped clean. They have the approval of all the war agencies—Army, Navy and Air Corps.

Double-shot molding is effective for unusual combinations of two materials of different colors. In this method the molded section of one color is utilized as an insert for the molding of the second. Provisions are made in the design for proper interlocking of the insert so that the two molded parts form an integral unit. This has been employed for decorative effects and also utility plates.

Special equipment has been developed to make simultaneous inking and indenting of printed detail into phenolic, urea or thermoplastic surfaces. Obviating hand filling of engraved or molded-in detail, the marking machine utilizes changeable hardened dies for characters or slugs. These when heated properly are impressed into the plastics to provide a permanent impression. The indentation and inking are performed in a 10-second cycle, the special ink drying immediately.

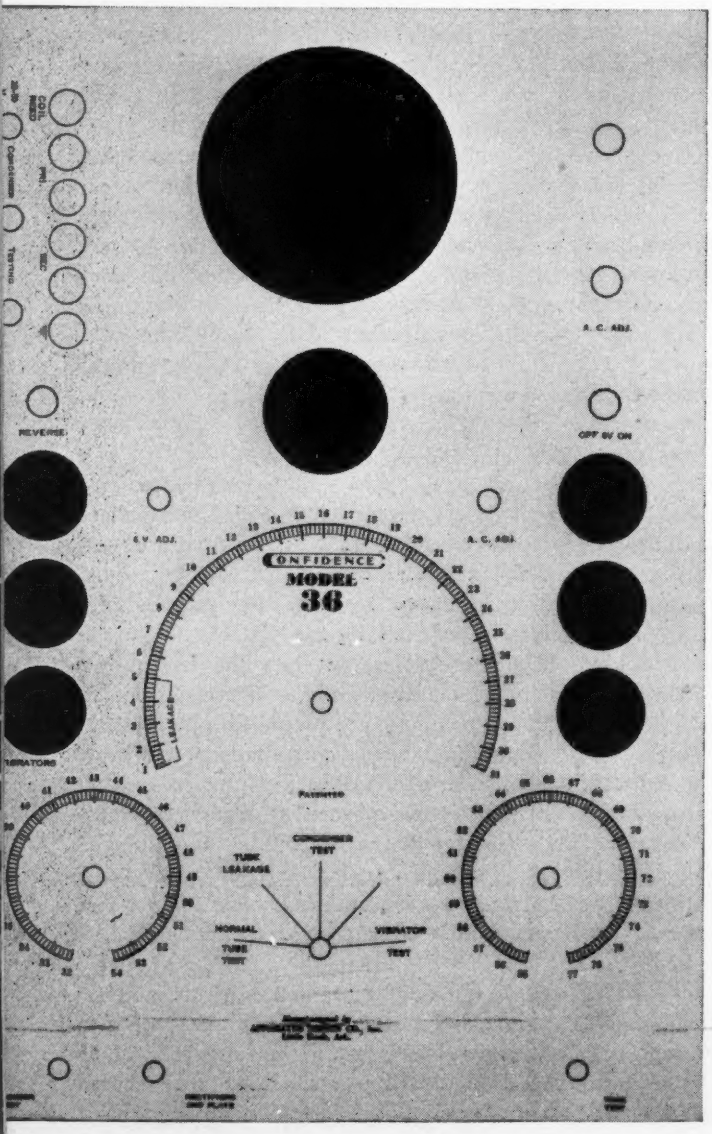


Fig. 9—Plate serves as mounting for instrument. Printed sheet is top lamination covered with a clear resin film

The recently announced technique of electroplating plastics might prove useful for depositing metal on molded plates, especially if a precious metal or critical metal is desired for deposition and if the light weight and other properties of plastics are useful.

Extruded cellulose acetate such as Tenite I and extruded cellulose acetate butyrate such as Tenite II are available in any shade of color and any degree of translucence, including clear transparent. Cellulose acetate has the disadvantage of getting brittle at low temperatures. It also expands and absorbs 7 per cent moisture. Therefore it has no dimensional stability. The material is readily available and plates, *Fig.* 4, can easily be produced.

The butyrate is suggested as preferable because of its greater dimensional stability, smaller likelihood of warpage, etc. Because it contains a smaller quantity of plasticizer for relative degrees of flow and is more compatible with plasticizers in general, there is less difficulty with filling or lettering pastes or lacquers. Unless the lacquer or paste is carefully selected there is a possibility that it will become tacky on continued contact with cellulose acetate, particularly in humid atmospheres.

Cellulose nitrate is a tougher material than cellulose acetate and is even more available. Its major disadvantage is that it burns at a higher rate. This, however, is not considered detrimental because the burning qualities are not pronounced when plates are fastened on a metal surface. These plates are particularly easy to manufacture.

Polyvinyls such as Vinylite are nonflammable, have dimensional stability, do not shrink or stretch and are tough. On large orders, however, it is difficult to obtain allocation from the Chemical Warfare Division of the Synthetic Rubber Section. Heretofore this division has taken the stand, where other plastics would do, not to allow use of this material. Scrap from other jobs nevertheless is available and small orders usually can be filled.

None of the now available plastics combine certain inherent advantages of metal plates, such as fire resistance, ductility, easy formability and good stamping properties. On the other hand certain plastics offer distinct advantages over metal plates, such as light weight, salt water resistance, insulating properties, and immunity to acids, alkalies or oxidizing agents. While ultimate availability of now critical metals will mean the return of metal plates, it is felt that a large number of present plastics applications will not revert to metals.

Helpful cooperation of the following companies is acknowledged for information and illustrations used in this article: The Burkhardt Co., Detroit; Chicago Molded Products Corp., Chicago; Eastern Etching & Mfg. Co., Chicopee, Mass.; The Formica Insulation Co., Cincinnati; General Electric Co., Schenectady, N. Y.; Mica Insulator Co., New York; Theodore Moss Co., Brooklyn, N. Y.; Parisian Novelty Co., Chicago; Tennessee Eastman Corp., Kingsport, Tenn.; Ward's Nameplates, Detroit; Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

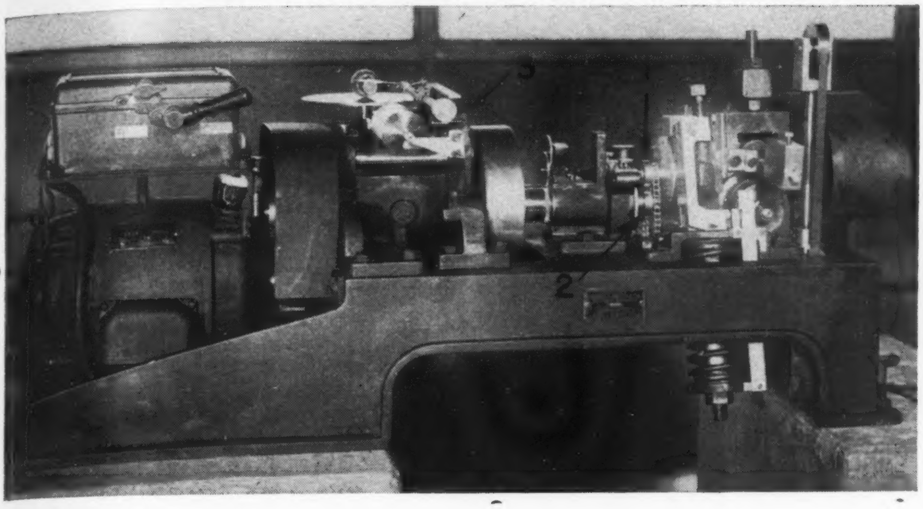


Fig. 23—Amsler wear-testing machine for seizure tests; 1 is specimen, 2 the shaft and 3 the torque recorder

Choosing the Right Material

Part IV—Bearings and Special Tests

By H. W. Gillett

War Metallurgy Committee

BEARINGS ARE an outstanding case where "standard" tests, aimed to appraise the properties of metals, wholly fail to measure those properties that really count in actual bearings. Nevertheless, bearing bronzes are still specified on the basis of the tensile properties at room temperature, determined on a test bar whose dimensions and rate of freezing on casting are unlike those of the bearing itself.

Compressive yield strength and—almost equally well—the hardness, at the bearing operating temperature are about the only properties determinable by "standard" testing methods that have direct meaning. Even these fall down in the case of a lined bearing where the thin lining is supported by a stronger back. Compression tests of massive metal of the same composition as this thin lining do not reveal the resistance of the backed-up lining.

Because bearing failures are disastrous and little understood, engineers are inclined to regard bearings with awe and to be reluctant to make any changes in a workable design. Tradition governs design to such an extent that substitution in the field of bearings is difficult. Any tests, regardless of their soundness, are viewed with suspicion and alternative bearing materials are not often tried on the basis of test results.

A bearing consists of the bearing itself, the oil or other lubricant, and the shaft or other coacting part. As long as there is oil between the parts, nothing happens. After a stop the oil film becomes more or less squeezed out, allowing metal-to-metal contact between minute high spots on shaft and bearing, except for any film of metal oxide, adsorbed oil or other material of that nature that may remain on these spots.

On starting, the relative motion tends to rub the spots clean

ONE OF a series, this article discusses the properties required of bearing materials in the light of recent developments and from the viewpoint of conserving strategic materials. Importance of proper testing and interpretation of results is stressed. General principles of special simulated-service testing of other materials are also covered in the article which is based upon a War Metallurgy Committee report for WPB

and to put them in condition to weld, i.e., to gall. If the bearing is too heavily loaded or the bearing metal unsuitable, the starting torque will be high, galling will start between high spots and damage will be done before the action of the bearing pulls in a complete film of oil. In heavily loaded bearings it sometimes is

necessary to lift the shaft off the bearing by oil pressure before starting.

Once the full liquid film of oil is established the bearing should run without trouble unless something happens to puncture the oil film, such as deflection of shaft or bearing under load due to lack of stiffness, to excessive loads, or to the entrance of a chunk of grit that cuts through the oil and oxide films and plows a furrow in the bearing. The quicker the grit is embedded below the surface of the bearing, the less the danger from this source but, if the grit sticks in without being covered, it acts as a lap. The bearing heats up in running, due to friction within the oil and due to any incipient seizure phenomena that may occur. Deformation due to thermal expansion sets in and, if the bearing is temperature-sensitive, it gets softer.

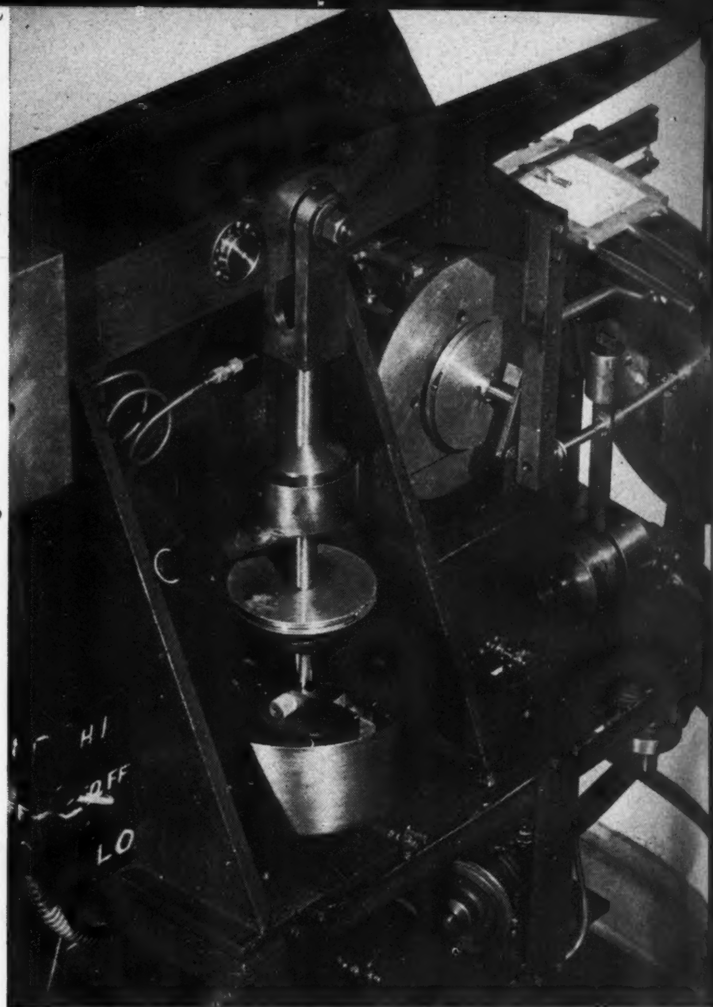


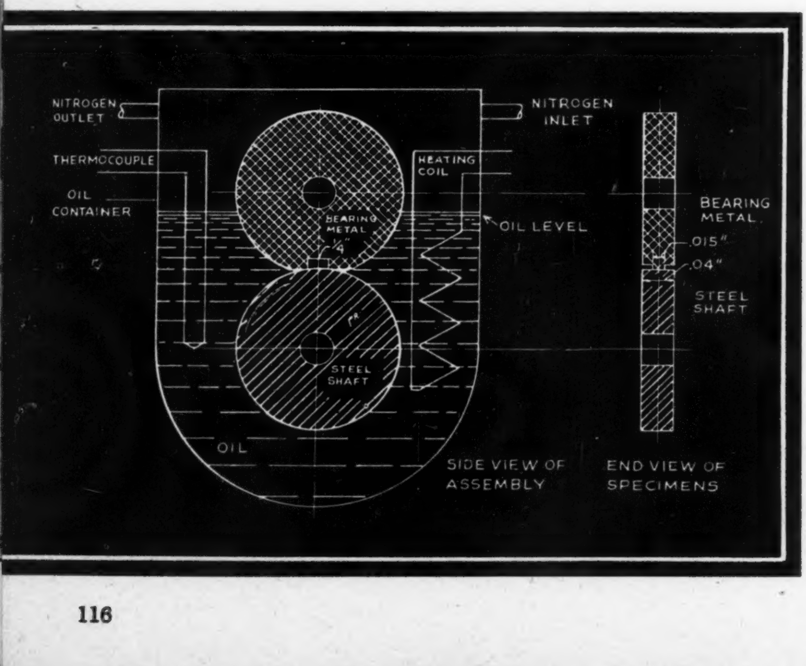
Fig. 25—Seizure tester in which pivoted-pad thrust bearings are used. Principle of operation is much like Amsler machine

Measuring Elusive Properties

On stopping, the oil film thins out as the bearing slows down and local contact and seizure phenomena are again favored. Plastic flow under pressure transmitted through the oil, or directly when the oil film is imperfect, may squeeze the bearing metal and allow it to become bell mouthed to accommodate shaft deflection. Excessive play in too large a clearance may exert a pounding action.

For materials under such conditions of service, the ordinary mechanical tests are of no value because none of them measure the elusive properties which make bearing metals different. Such properties can be measured only by running a material as a bearing and finding its behavior under load. Tests of bearing materials in this way show that excessive loads cause failure by seizure, which results from high friction leading to excessive heating. The reason for the high seizure resistance of good bearing materials may be due to a high capacity to adsorb an oil film, or to acquire an oxide or other nonwelding film (beneficial dirt) or to some other property not apparent. At any rate, the only way to evaluate this property—which is perhaps the only one which might properly be called a "bearing" property—is to test it in a bearing.

Fig. 24—Below—Detail of specimens used in the Amsler machine of Fig. 23. Bearing specimen can be smaller than shown and does not rotate in seizure test. Load is increased in gradual stages until the coefficient of friction rises sharply, indicating the region of incipient seizure



Mechanical properties needed in a bearing include: An adequate compressive yield strength at operating temperature to resist squeezing out under the bearing load, yet a yield strength low enough, if possible, to embed a grit particle. A low modulus of elasticity, so as to allow considerable elastic deflection back of a piece of grit or to accommodate shaft deflection, may be an asset. Fatigue resistance is needed for intermittently loaded bearings which often fail in fatigue. Cracking occurring on a lined bearing may be due to lack of fatigue resistance of the lining or to imperfect bond between shell and lining. Notched-bar impact resistance might possibly be a feature in a loose bearing with improper oil grooves, but the service does not allow deformation of the order produced in an impact test to fracture.

Modern babbitt-lined bearings have thin linings for reasons analogous to those that demand thinness in soldered joints. The support given by the backing makes a thin lining stronger than

a massive specimen of the lining material. With precision machining so that a bearing will finish up without cutting through to the backing, and with rigidity in shaft and bearing, it is possible to make the lining thin enough so that this added strength is utilized. By such means the automobile engine is adequately served by lead-base babbitts. In massive form or in thick linings, they would not have a high enough compressive yield strength to resist squashing out but, in the thickness used, they are at least as satisfactory as were the thicker, tin-base babbitts previously employed, even though lead-base babbitts get so soft at operating temperatures that they were not suitable when thick linings were in vogue.

A committee of the Society of Automotive Engineers says[1], "Lead-base babbitt bearing can be expected to show mileage superior to that of tin-base *providing* the babbitt thickness does not exceed .035-inch. Between .035 and .06-inch the performance can be expected to be on a par. If the lead-base babbitt is over .06-inch thick, its performance will probably be inferior unless the severity of operation is reduced."

A candidate alloy, aspiring to be a "bearing metal", can be given a preliminary examination as to hardness at room and at bearing operating temperature. In the case of lead-base alloys where rate of application of load influences strength and where precipitation and aging tendencies might be present, slow loading in the hardness test, after proper aging, is necessary. The compressive yield strength on massive specimens can be determined directly or similar information obtained more conveniently by the repeated pounding test developed at the Bureau of Standards[2] than by regular compression testing at elevated temperatures. The specimen for this, though small, is still massive as compared with a thickness of modern bearing lining. Some idea of resistance to squashing and of embedability can be had from such tests.

Testing Seizure Characteristics

Summing up, propensity toward seizure has to be measured by some test in which the material of the shaft, with the finish to be used on the shaft, is rubbed against the material of the bearing, with the finish it is to have, in the presence of the lubricant to be used and, to complete the story, in the presence of representative grit. Since bearings operate at an elevated temperature, provision must be made for control of temperature. A convenient seizure test can be made on the Amsler machine shown in *Figs.* 23 and 24. Another device is shown in *Fig.* 25.

The regulation endurance test, does not appear to give information of much value from massive specimens. A type of repeated flexure test that can be run on bushings or on lined bearings is illustrated in *Figs.* 26 and 27.

If tests indicate that a new alloy is not too obviously lacking in some important feature of behavior, it can be examined in simulated service, employing actual thickness, finish, shaft clearance, etc. The General Motors machine, *Fig.* 28, though specially designed for testing connecting rod bearings, can give information—especially with respect to fatigue resistance—which is useful on other bearings. An actual bearing can be tested for behavior, particularly as to seizure, as shown in *Fig.* 29. Some idea of the behavior of the alloy as to corrosion by active oil can be had from special tests, such as the Underwood test, but such tests are notorious for failing to place materials in the proper order if any deviation is made from the exact type of oil and the exact temperature of service.

Even all these tests put together do not guarantee that the alloy is entirely suitable for a given installation, especially if the exact features of stiffness of shaft and bearing in the actual design have not been duplicated. Distortions that may occur through the particular temperature distribution in the actual design are unlikely to be duplicated in a test setup. Cruel and unusual punishment may be present in actual service. But, as far as a bearing material can be evaluated prior to actual service, tests can at least indicate whether the material is unsuitable or has a chance to perform satisfactorily.

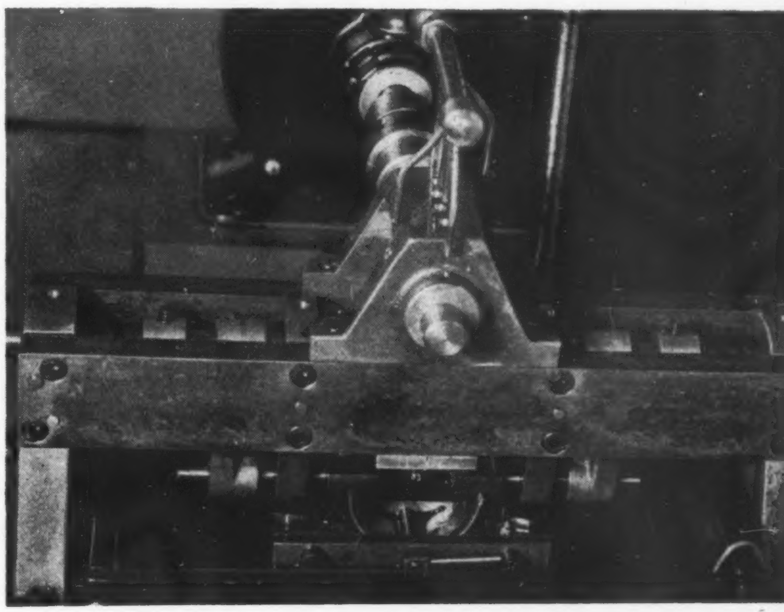


Fig. 26—Test for fatigue of a lined bearing half shell. Ends are flexed .003-inch by a reciprocating device

By such tests, and by actual service, a variety of bushing alloys and bearing linings have been developed. In attempting to generalize on the metallurgical requirements the superstition grew up that a bearing must have hard particles in a soft matrix or soft particles in a hard matrix and that a uniform, homogeneous metal or alloy could not act as a bearing. This is a backhanded way of saying that depressions, such as are produced by wearing in when the bearing has hard particles in a soft matrix, are good oil reservoirs but it neglects the obvious fact that the depressions could be produced mechanically. There are strong advocates of intentionally pockmarked bearings. It also neglects the fact that, within their limited load-carrying ability, massive bearings of pure lead have ex-

[1] "Report to Office of Defense Transportation on Engine Bearings", Society of Automotive Engineers, 1943, Page 3.

[2] French, Rosenberg, Harbaugh, and Cross—"Wear and Mechanical Properties of Railroad Bronzes at Different Temperatures", *Bureau of Standards Journal of Research*, Vol. 1, (1928). Pages 343-421.

cellent behavior and that pure silver bearings coated with pure lead are among those best suited for severe duty, the load-carrying ability of the lead film being vastly raised by its stronger backing.

Another thing that is disregarded when the usual generalizations are expressed as to soft pools in a hard matrix is that these soft pools are generally lead and provide a smear of lead over the surface. Outside of the tin-base babbitts (which contain a smearable matrix) and some extremely hard, high-tin, nonleaded bronze bushings, nearly all good sleeve bearing metals contain lead, cadmium or some soft metal that smears as lead does.

Tin-base babbitts, tin hardened with antimony and copper, were the standard of excellence when thick babbitt linings were in vogue, though lead-antimony and lead-antimony with a little tin were always used in larger tonnage than the tin-base alloys and were generally adequate for those cases where the operating temperature was not too high. The lead alloys lose compressive yield strength and hardness as the temperature rises, while the tin-base alloys do not soften to such a degree.

In thin linings, the regular, oldtime, lead-base babbitts act as well as did the tin-base babbitts in thick linings, except for resistance to corrosion by some active oils at elevated temperatures. Cadmium-base babbitts then came to the fore. They were able to stand somewhat more severe duty than even the tin-base babbitts and their hardness temperature curve only fell off slowly with temperature, much as the tin-base babbitts do. The corrosion resistance of cadmium-base alloys to active oil at high temperature is much poorer than that of the usual lead-base alloys, so, while the cadmium alloys surpass tin-base in load-carrying ability, they are inferior in those cases where neither excessive temperature nor active oil can be avoided. The cadmium-base babbitts would serve widely in place of tin base, were there enough cadmium to meet the demand, but there is not.

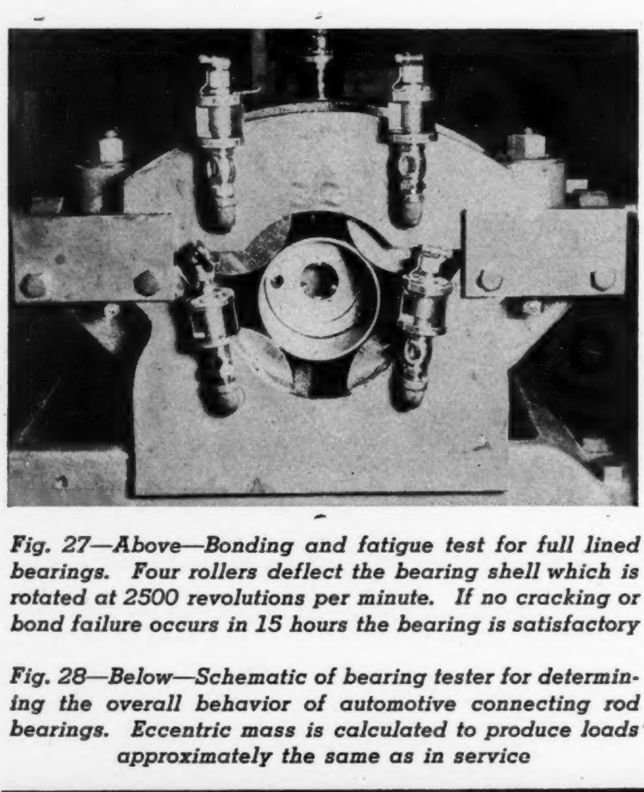


Fig. 27—Above—Bonding and fatigue test for full lined bearings. Four rollers deflect the bearing shell which is rotated at 2500 revolutions per minute. If no cracking or bond failure occurs in 15 hours the bearing is satisfactory

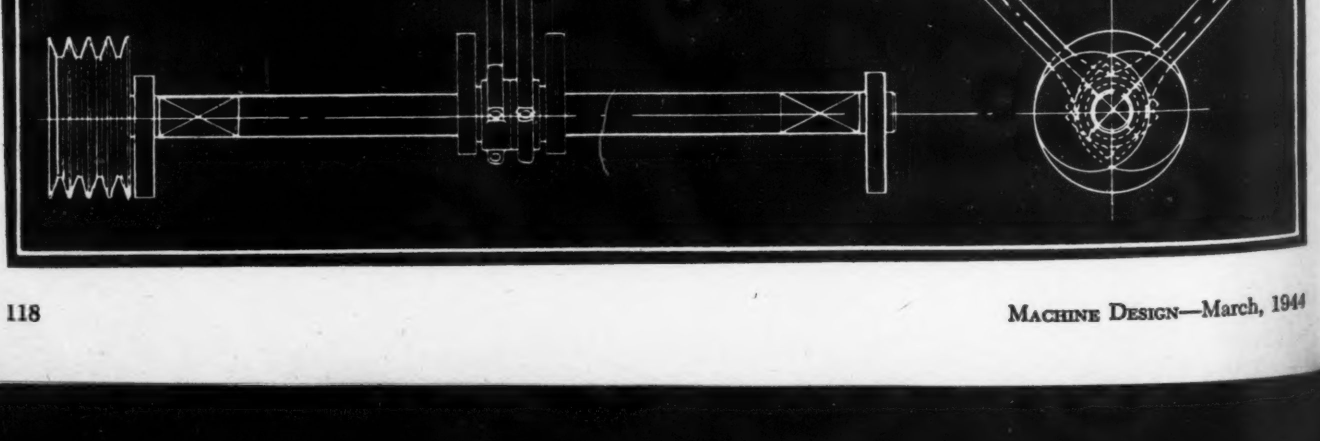
Fig. 28—Below—Schematic of bearing tester for determining the overall behavior of automotive connecting rod bearings. Eccentric mass is calculated to produce loads approximately the same as in service

Modification of the usual lead-base babbitts has taken two routes. The older one employs calcium as a hardener, plus a small amount of tin, and other elements for further hardening or for improving resistance to active oil. The corrosion resistance is not, however, sufficiently improved. Necessary control of the calcium content brings in a production problem such that skill and experience are needed. The alloy does not have as favorable "soldering" or "tinning" behavior as some others, and the relatively poor corrosion resistance limits it to rather low temperature use, but it does serve as a limited replacement for tin-base babbitt.

The other modifications retain antimony as the main hardener, use a little tin and introduce either arsenic or silver to bestow better hardness at elevated temperatures. Although former specifications for lead-base babbitts limited arsenic to .25 per cent or less because of alleged brittleness and injury to bonding properties, one alloy uses about 82½ lead, 15 antimony, 1 tin, 1 arsenic and .5 copper to duplicate the behavior of average tin-base babbitts, and may even have still higher antimony content.

Another combines the soldering properties of the lead-silver alloys with the bearing properties of the lead-antimony alloys and balances the arsenic and tin contents so as to produce satisfactory resistance to corrosion by active oil. This contains 10 to 15 antimony, 2 to 3 tin, 2 to 5 silver, and up to .25 per cent copper, depending on the grade of tin-base babbitt—hard, medium or soft—it is desired to duplicate. It has good bonding properties and a hardness-temperature curve coinciding closely with that of the grade of tin-base babbitt being duplicated. These alloys use much less tin than does the common lead-base babbitt of 75 lead, 15 antimony, 10 per cent tin in which the tin is kept high primarily for bonding purposes.

The lower-tin, common lead-base babbitts, the calcium-



hardened or arsenic-hardened types, may need a "tinning" of the bearing shell before applying the lining metal. For such "tinning" the 96 lead, 4 per cent silver solder is useful.

There is little need for tin-base, cadmium-base or specially alloyed lead-base babbitts when extremely thin linings are used, for lead containing 10 to 15 antimony, 2 to 5 tin, and up to .25 per cent copper, applied with good practice as to fluxing and "tinning", will do about all that any babbitt will. It is in the cases where thick linings have to be employed, as for machining tolerances or relining old bearings, that the special babbitts fill a need. Practically all the needs formerly served by tin-base babbitts can be served by babbitts with not over 5 per cent tin.

For service too severe for babbitts, recourse is commonly made to "copper leads"—a group of alloys with a matrix of copper or copper with a small amount of tin, sometimes with a little silver or nickel instead of or together with the tin, carrying 20 to 35 per cent lead distributed through the matrix in pools. The lead smears over the running surface, and the pools, from which it comes, wear down or corrode to provide little oil-holding depressions. The matrix itself has none too good resistance to galling, the alloys will not stand starved lubrication as well as the babbitts, do not wear in rapidly, require hard shafts, need increased clearance and must have uninterrupted lubrication, as compared with babbitt. Securing uniform distribution of lead in tiny pools instead of in large gobs is a headache to the producer. The cost of a good copper-lead bearing is all out of proportion to the cost of the raw materials because of the fussy control that has to be exercised with respect to lead distribution and bonding.

In spite of these drawbacks and of the danger that the lead will be corroded by active oil, the higher compressive strength and especially the load-carrying ability at elevated temperature of the copper-leads over babbitt have led to their wide use. One saving grace is that, when a bearing does fail, there is some chance that enough lead will melt out, with the matrix staying put, to prevent as much damage as when matrix and all melt out, as with babbitt.

Silver Proves Excellent

For very severe service, as in aircraft bearings, silver is coming into important use. It has a fairly low elastic modulus, about 10 million, and is very fatigue resistant. Its hardness, about 25 Brinell, is not too high to afford some embedability. Indeed it is within the range of that of the stronger babbitts at room temperature it maintains this hardness at bearing operating temperatures and it is fairly seizure resistant. A little lead greatly improves its seizure resistance but injures its bonding properties when cast on a steel back. Using pure silver to secure bonding and electroplating a film of lead over the bearing face provides insurance against seizure as long as the lead is not removed by oil corrosion. To avert corrosion, a still thinner film of metallic indium is sometimes plated over the lead. Another promising scheme is to make the bearing face by electrolytically codepositing silver plus a few per cent of lead. In fact, for linings of the thinness favored at present, making the whole lining by electrodeposition offers some advantages

"Sandwich" bearing linings are in use in which a porous layer of bronze, copper-nickel or the like is adhered to the steel back by powder metallurgy methods. Then the voids are filled and the bearing surface provided with a lead-base babbitt. This scheme builds up a "copper-lead" type of structure, avoiding some of the bonding difficulties of the fused copper lead, and the babbitt pools are much more resistant to corrosion by active oil than is the lead of the copper-lead type. Thus, for lined bearings, the engineer has a number of choices among excellent materials that have working properties previously afforded by the tin-base babbitts.

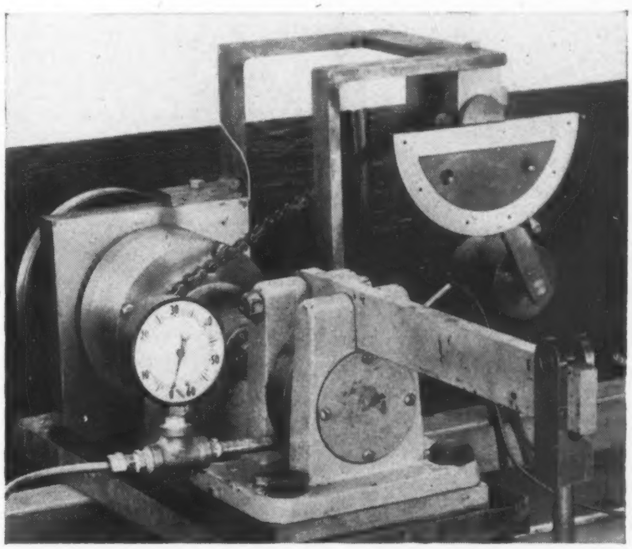


Fig. 29—Sleeve bearing testing machine for seizure studies. Load-carrying ability is measured by maximum load which can be supported without excessive friction

In unlined bearings, i.e., bushings, a copper-alloy matrix carrying lead pools, 5 to 15 per cent, is the regulation material. These have high compressive strength as compared with the copper-lead lining of higher lead content and can be cast easily with uniform distribution of the lead. The composition most used is 80 copper, 10 tin and 10 lead; though 85 copper, 5 tin, 5 zinc and 5 lead is often found to serve. On the whole, however, too much zinc is considered to be detrimental to seizure properties and a lead content higher than 5 per cent is preferred, so the tendency has been to cling to the 80-10-10 composition. When that composition is demanded and the "impurity" limits are low, only scrap of 80-10-10 composition can be used. Available scrap of other types cannot be worked in, so the requirement for new tin is high. It would also be helpful if the requirement for new copper could be kept low.

The extent of real engineering need for the high compressive strength of 80-10-10 has been questioned by Etchells and Underwood[8] who point out that if a plain bearing has to start under load, the permissible load is limited by the permissible starting torque so that, unless some means is employed for lifting the journal from the

(*Continued on Page* 174)

[8] Etchells, E. B., and A. F. Underwood—"Practical Aspects of Bearing Design", MACHINE DESIGN, Sept., 1942.

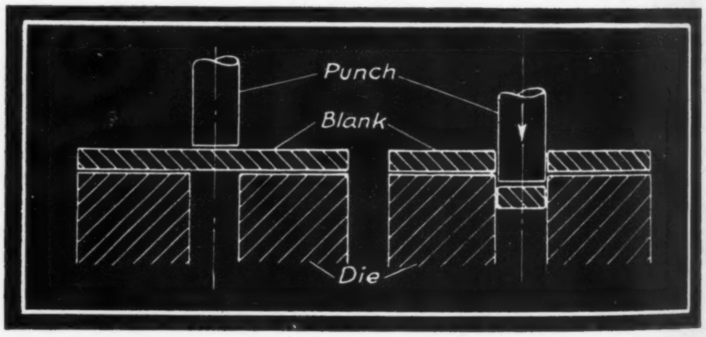
Redesigning To Utilize Stamping Process

Part II

By Colin Carmichael

Fig. 7—Above—Pyrotechnic pistol, originally made primarily from machined castings (left), now is composed largely of steel stampings (right)

Fig. 8—Below—Basic elements of the punching operation. Addition of stripper plate on top of blank facilitates subsequent withdrawal of punch



COMPLETE REDESIGN of a self-contained machine—the 60-inch Army searchlight—to utilize stampings wherever possible was discussed in the previous article. As an indication of what can be done on a smaller scale, the redesigned pyrotechnic pistol illustrated in *Fig*. 7 affords an interesting example. The original pistol, made primarily from machined castings, cost $34.00 complete with mount. The new design, utilizing steel stampings extensively, cost $11.00 and weighed 40 per cent less. Much of the saving in cost is due to the adaptability of the stamping process to low-cost mass production.

Shown in the upper part of *Fig*. 7, the redesigned pistol, which is made by the Eureka Vacuum Cleaner Co., uses the same shells as the original design shown beside it. Upper half of the frame is enclosed between two stampings held in place by screws. A pair of thin steel stampings or liner plates, fitting over the pivot pins, provide tabs to space apart and guide the working parts of the firing mechanism. The barrel hinge, breech-lock housing and four recoil lugs—also stampings—are spotwelded and silver soldered to the barrel. Rear end of the barrel is notched on the press for the ejector. The recoil mount, which is built up of steel stampings and welded steel tubing, weighs 50 per cent less than the original mount.

Because one of two things may happen when a blank is pressed between a punch and die—see *Figs*. 8 and 9—it is important to design the part so that the desired result, and only the desired result, shall occur. Thus, in forming a deep-drawn cup, *Fig*. 9, provision of adequate radius on both punch and die, as well as the proper clearance, obviates any possibility of the bottom of the cup shearing out, as in *Fig*. 8, inasmuch as the shearing or punching action depends on sharp edges. Inside radius of a drawn part therefore should exceed the thickness of the stock. Outside radii usually are considered as being greater than the inside radii by the thickness of the stock.

Simplest shape of a drawn part is the cup, *Fig*. 9. Modification of this basic form will be recognized in most of the parts illustrated in this article, including *Figs*. 10, 11 and 12. Flanged and dished heads, of which the part shown in *Fig*. 10 is a modification, are available in certain standard sizes and shapes. Where it is possible to work these particular shapes into a design, savings may be effected through the distribution of die costs, particularly when the number of parts required on one order is small. The part shown in *Fig*. 11 is an accumulator housing for an aircraft hydraulic system and is typical of the kind of part which is particularly well adapted to the deep-drawing process. With a diameter of 8 inches, the housing is designed for a working pressure of 1000 pounds per square inch.

Drawn stampings welded to form an important ordnance part are illustrated in *Fig.* 12. This is the top roller for a half-track, which formerly was a machined steel casting costing $15.00. The redesigned roller, as developed by the Cleveland Welding & Manufacturing Co. in cooperation with engineers of the White Motor Co., now costs only $6.50. The casting weighed 42 pounds and required 60 minutes machining time whereas the stampings are produced from 25 pounds of sheet steel and the machining of the weldment requires only 20 minutes, a saving of 17 pounds of steel and 40 minutes machining time.

A brazed assembly of formed parts for another ordnance component is shown in *Fig.* 13. When machined from bar stock this front leg hinge for a .30-caliber machine gun tripod mount used 1½ pounds of bar which required two minutes machining time. Stampings require only ½-pound of strip steel and no machining is needed. A joint development of the Cleveland Ordnance District and the Toledo Stamping and Manufacturing Co., the hinge was reduced in cost from 50 cents to 18 cents.

Strains and Stresses Due to Forming

Severity of the deformation that occurs in the forming of parts such as those just discussed places certain restrictions on the materials that can be so formed. It will be noted, for example, that in *Fig.* 9 the circumference of the drawn cup is considerably less than that of the flat blank from which it was made. Reduction of the circumference involves a severe tangential compressive stress in the blank and flange during drawing, while the force of the punch induces high radial tensile stresses. The blank holder is intended to prevent the formation of wrinkles due to the compressive stresses. Once formed, such wrinkles could not be removed. To permit flow of the metal under these stresses it is necessary to work in the range between the yield point and the ultimate strength

Fig. 9—Below—Drawing operation depends on provision of proper clearance and radii. Blank holder may be omitted if blank is relatively thick

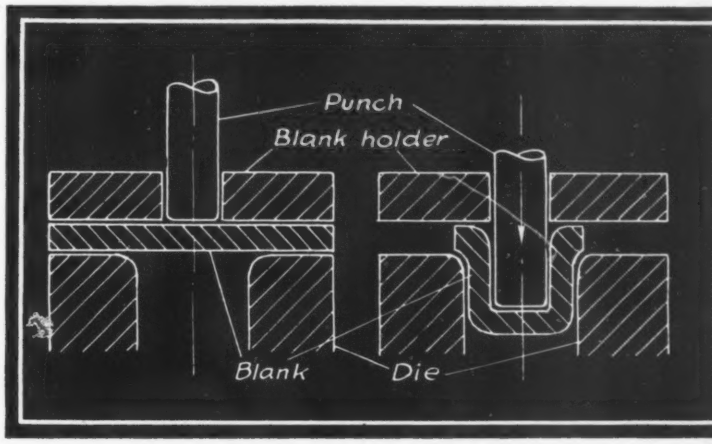


Fig. 10—Below—Circular drawn shapes of this type are well adapted to production in relatively small lots

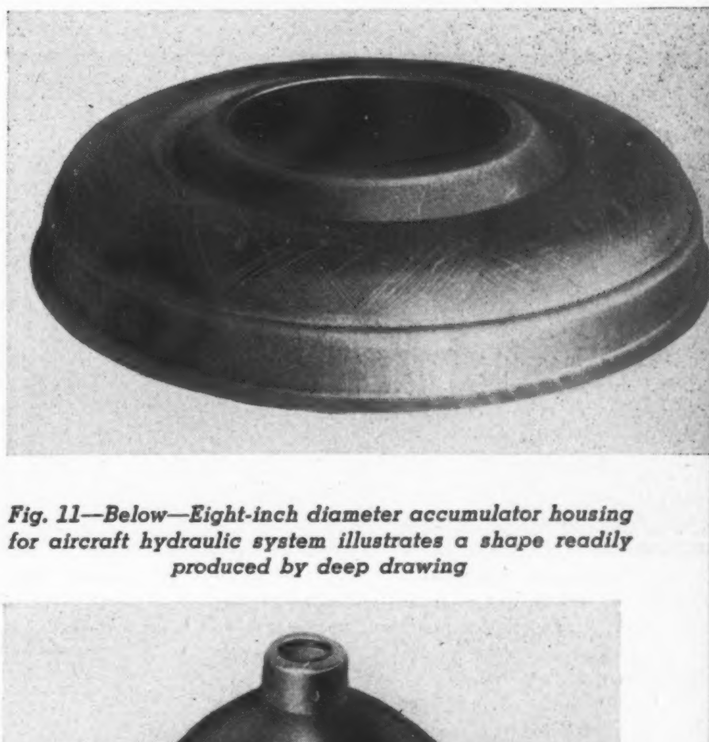


Fig. 11—Below—Eight-inch diameter accumulator housing for aircraft hydraulic system illustrates a shape readily produced by deep drawing

of the material; therefore if severe deformation is involved, a material with a large spread between these points should be chosen.

When a metal is worked cold, the deformation occurs wholly on slip planes within the crystals. Below a certain temperature, called the recrystallization temperature, the distorted grain fragments cannot rearrange themselves into their normal shape, and the material becomes progressively harder and less ductile as the cold work continues. This work hardening is beneficial up to a point, because the deformed part becomes stronger and resists further yielding, thus distributing the deformation and preventing excessive local yielding at the points initially stressed. The limit of cold work is reached when any part strain hardens to the breaking point. Reheating to the recrystallization temperature, known as process annealing, restores the ductility and permits further cold work. Because annealing is expensive, the designer should if possible so proportion the part and select the material that forming can be completed without annealing.

In evaluating a material for deep drawing, prime considerations are the ratio of yield to ultimate strength in tension, the hardness, the per cent elongation, and the reduction of area. Grain size, uniformity of grain size and direction of grain also affect the quality of the work, particularly the surface finish. Oversize grains cause a rough surface in highly strained material, which is descriptively termed the "orange peel" effect. A fine-grained material successfully withstands much greater deformation. Also of importance is the shape of the tensile stress-strain diagram; if there is a pronounced dip at the yield point, as in the case of certain soft annealed materials, "stretcher strains" may appear on the surface during deep drawing. This condition is avoided if the annealed sheet or strip is given a slight amount of cold rolling prior to drawing.

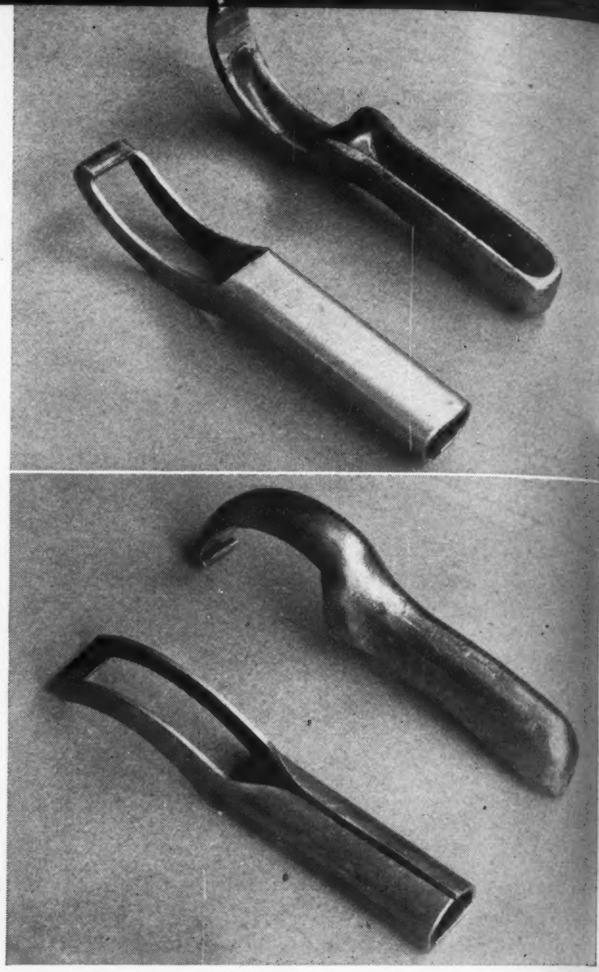


Fig. 14—Upper piece in each view shows casting of a draw-spindle handle for a bench lathe. Lower piece is the part redesigned as a stamping

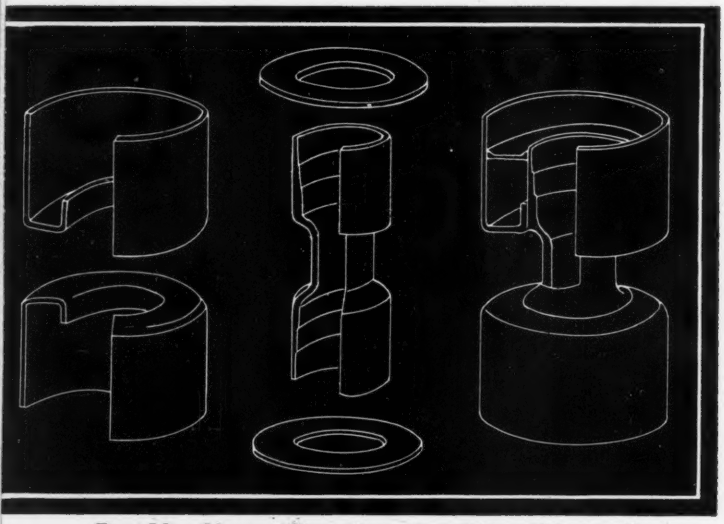
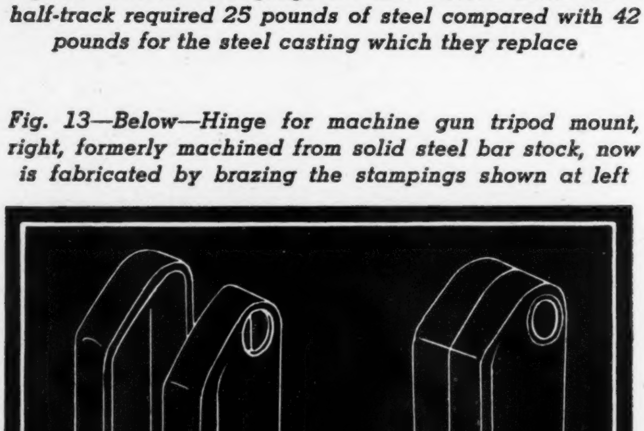


Fig. 12—Above—Stampings for this welded roller on a half-track required 25 pounds of steel compared with 42 pounds for the steel casting which they replace

Fig. 13—Below—Hinge for machine gun tripod mount, right, formerly machined from solid steel bar stock, now is fabricated by brazing the stampings shown at left



Most materials can be drawn and formed, although steel, brass and aluminum are the most widely used. For severe work in steel, carbon content under .35 is preferable, a popular grade being an open-hearth steel with .05 to .08 per cent carbon and .25 to .50 per cent manganese. Machining steels containing sulphur are highly undesirable. Both strip and sheet steel are available either hot or cold rolled, the cold-rolled material being superior in accuracy and appearance. Amount of cold rolling is designated by the difference in thickness gage numbers before and after, the reduction in thickness in going from one gage number to the next being a constant percentage in any system. Ordinarily sheet is cold rolled only one number whereas strip may be rolled two to four numbers, giving a harder material. Tempers of mild steel also are designated by descriptive terms such as dead soft, soft, ¼-hard, ½-hard, ¾-hard, hard, extra hard, spring hard, etc. While thicknesses often are designated in gage sizes

it is always safest to specify actual decimal thickness inasmuch as several gage systems are in common use.

Stainless steels in the soft tempers can be deep drawn and formed with less need for annealing than mild steel, but require more power and cause more die wear.

Because of their high ductility, certain of the brass alloys are highly satisfactory for deep drawing. For many purposes their good corrosion resistance, pleasing appearance and other desirable characteristics are an additional

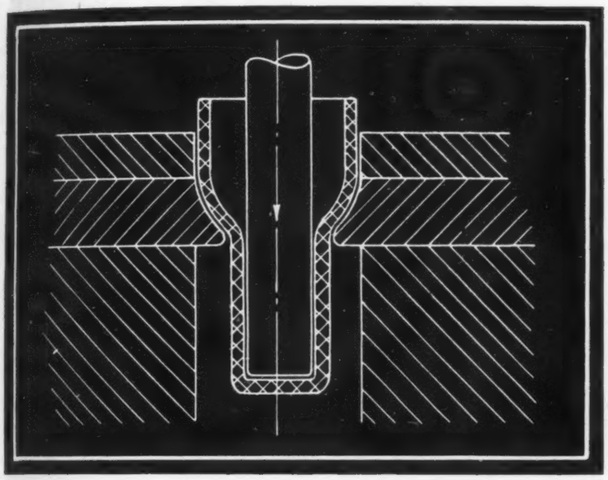
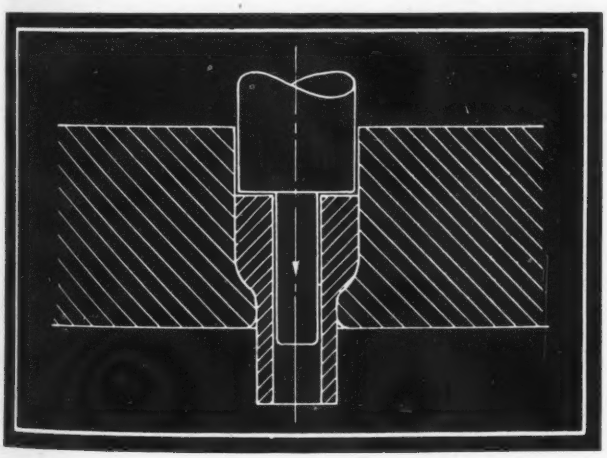


Fig. 15—Above—Ironing is used to reduce drawn shells

Fig. 16—Below—Extrusion reduces wall thickness by squeezing material through space between punch and die



recommendation, although the current materials situation necessarily limits their use at the present time. Standard for deep-drawing is cartridge brass (70-30 copper-zinc), while tobin bronze, phosphor bronze and other brasses with somewhat less ductility also are used.

Pressed aluminum parts are finding extensive use today, particularly in aircraft. Alloys 2S, 3S, 4S, 24S, 52S, 53S and 61S all are well adapted to forming. For general work the softer, more ductile alloys such as the first three listed are used if their strength is adequate. The heat-treatable and age-hardening alloys must be worked within a short time after annealing, then heat treated.

Of the nonmetals, thermoplastic sheet already is being deep drawn and formed on a commercial scale. Vulcanized fiber, rubber, plywood, impregnated papers and fabrics, etc., can be worked by modifications of the stamping process.

Of particular interest to the designer of machines which are not built in large quantities is the use of so-called "temporary" dies. With only 15 or 20 per cent of the cost of ordinary permanent dies, these are said to give just as good results where the total run is not more than 5000. Actually, many parts with a total run as low as 500 are made in such dies, the cost comparing favorably with other methods of fabrication. The draw-spindle handle, *Fig.* 14, used for tightening the handwheel of the draw-in spindle for collets on a bench lathe is an excellent example of a part required only in small lots but which can economically be formed in such temporary dies.

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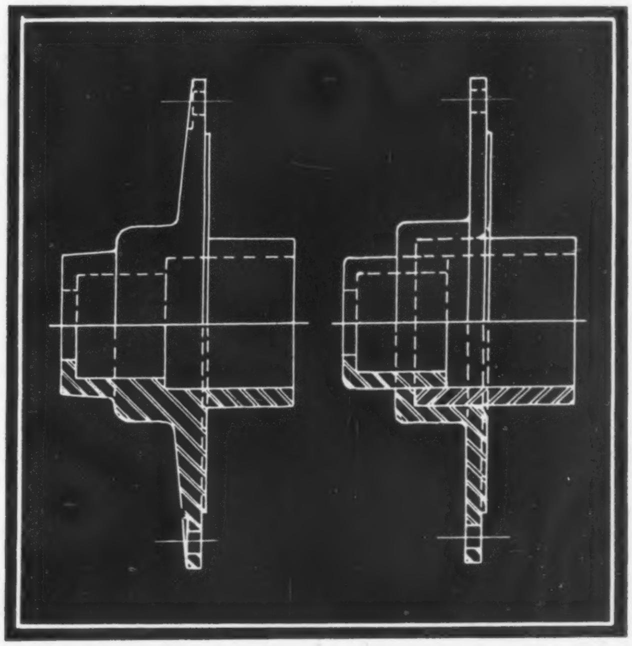
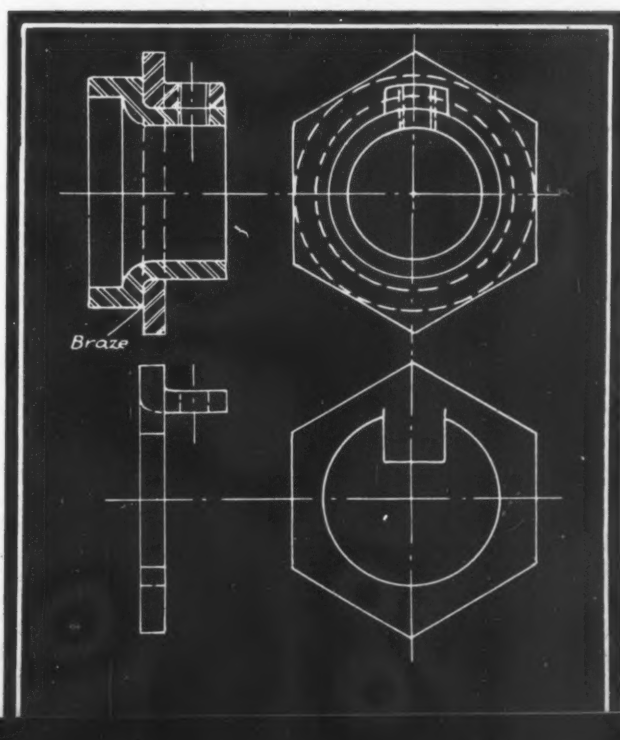


Fig. 17—Above—Comparative designs for a hub machined from a forging, left, and an assembly of brazed stampings, sized and coined, right

Fig. 18—Below—Hexagon wrench grip and boss for set-screw are formed in one piece from 3/16-inch sheet metal, then brazed to seamless steel tubing



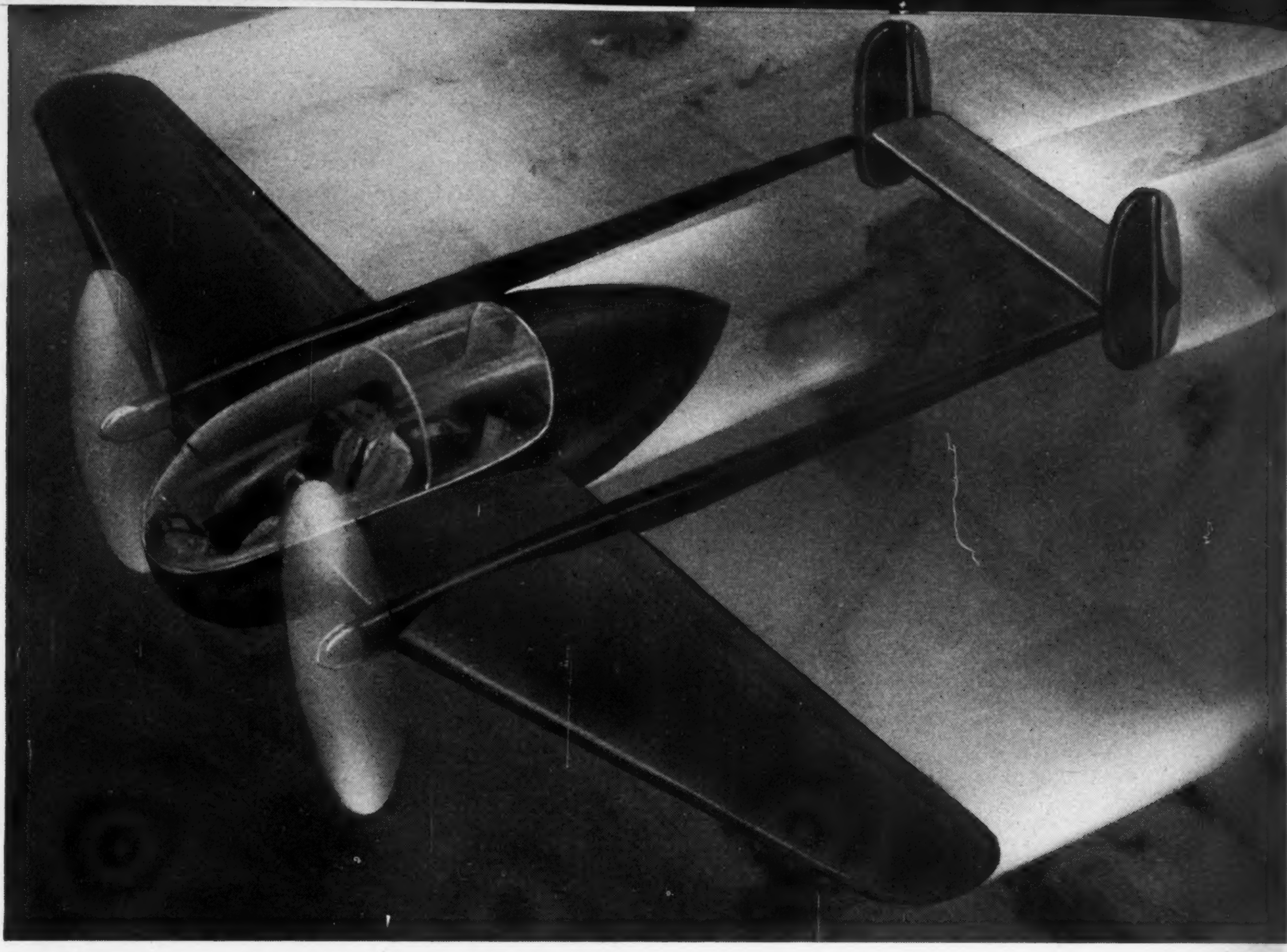


Fig. 1—Will the private plane of the future look like this? It is visualized that such a four-place job might be powered by two 75 brake-horsepower internal combustion turbines. Top speed: 165 miles per hour. Landing speed: 50 miles per hour

Forecasting the Private Plane of the Future

By William D. Hall

Chief Engineer

Aeronca Aircraft Corporation

Reasonably soon after the war's termination, the individual will have a choice of aircraft for his private use which will be vastly superior to that to which he has been accustomed in the past. Although immediate wartime developments have not been concerned with the private type aircraft, the explosive rate at which the aircraft industry has progressed has brought to light many improvements which will be applied to the private airplane. These developments, together with those occurring naturally in a field of intense competition —most aviation leaders feel that the private plane postwar competition will be terrific—are bound to result in a number of excellent models for the public use.

At present, we in the aircraft industry are able to do only a limited amount of development on the future's aircraft. However, even if we could completely develop a design, there would not be sufficient personnel to tool a factory. Nor for that matter, could we obtain the materials with which to start production the moment a "go ahead" is given. Hence, the immediate postwar market will be supplied with ships which are identical or only slightly modified from those offered in 1940 and early 1941.

Although these ships can take many forms, the important ones are the "fixed-wing" type and the "rotating-wing" type such as the helicopter. Considerable publicity has been given the recent advances of the helicopter, but I feel they are still a long way from offering the best compromise from the standpoint of the private owner. Apparently then, the fixed-wing type will dominate the immediate future and it is primarily this type which will be discussed in the following.

Judging from present trends and indications, it appears relatively certain that at least the three types listed below will be offered to the postwar public:

1. A low-powered, inexpensive training type aircraft,

Abstract of a paper presented at the recent War Engineering annual meeting of the Society of Automotive Engineers in Detroit.

with either tandem or side-by-side seating arrangement, which will be a refined version of the prewar light airplane models. This airplane will sell at approximately one thousand dollars, and in order to meet this cost, will consist of the minimum amount of aircraft that will satisfactorily fly.

2. A medium priced, medium performance type aircraft which will be super-safe. Aimed at the private owner, it will be available in two-place and four or five-place models. The two-place version will list at approximately fifteen hundred dollars and the larger ships at about three thousand. Performance of this type of ship will be compromised to some extent to obtain maximum safety and comfort. As the development and tooling costs are amortized the price will move downward, but probably will not reach the desirable one thousand dollar level for the two-place, nor the two thousand dollar level for the larger ships for some time.

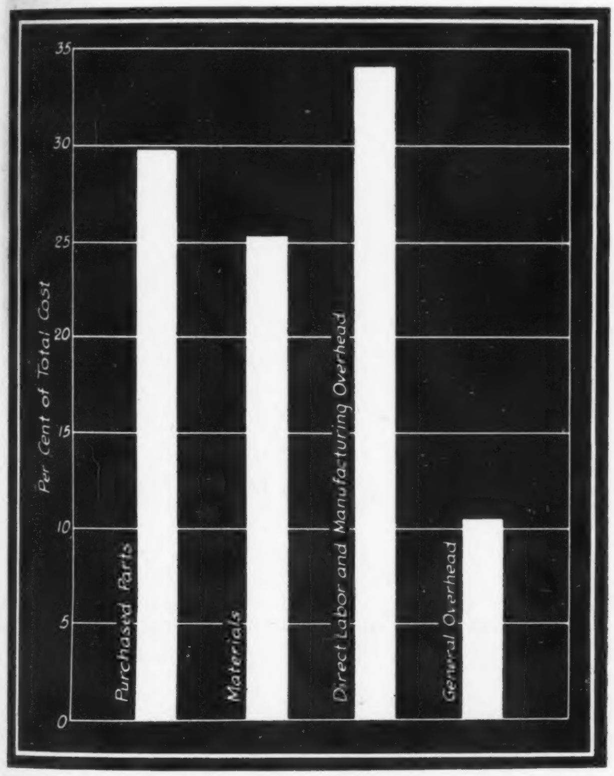


Fig. 2—Major obstacle to price reduction of aircraft is the high cost of purchased parts and materials. Chart shows that such items—beyond the plane producer's control—constitute well over half of the total manufacturing cost

3. An aircraft offering performance as high as is consistent with reasonable safety, comfort, and cost. It will be available at approximately two thousand dollars. This aircraft will be a "cleaned up" version of the prewar two-place, side-by-side, low-wing, retractable-gear type, and will be offered to those whose demand is for maximum performance.

Standards of design, accepted heretofore in private aircraft, will be superseded by much higher ones. It is only by these higher standards that the inherent difficulties of operating aircraft will be minimized to a point where the average person can easily handle one. The ideal airplane will be that one which is the best compromise between safety, cost, performance, dependability and comfort.

Before anyone will accept something new, he satisfies himself that neither he nor his family will encounter any greater risks than they are accustomed to. Therefore it will be necessary to develop the public's airplanes so that they will feel no more uncomfortable in them, as far as personal hazards go, than when they are in their automobile or bathtub.

Overcoming the Stall and Spin

It has been found that the primary form of loss of control and stability is the stall followed by the spin. Spin-proof or spin-resistant designs in the immediate prewar period offered at least a partial solution, and it is reasonable to expect further refinements through research which will be undertaken after the war.

Nonspinning characteristics of the future will probably be accomplished as they were in the past, by the following means: The wing proper will be designed so that it stalls gradually with the stall originating over the center portion of the wing, and the ailerons used will not stall the remainder of the wing when they are deflected to obtain lateral control. Further, longitudinal control will be limited so that it will be impossible to place the airplane in a complete stall position without a dive and a violent pull-up. Also, an automatic warning of the impending stall, in the form of tail buffeting, or perhaps an instrument on the dash board will be included for the benefit of the individual who might occasionally become careless.

When a person suddenly takes up flying, he finds it extremely difficult to coordinate the use of the rudder and aileron. Most individuals master this operation after

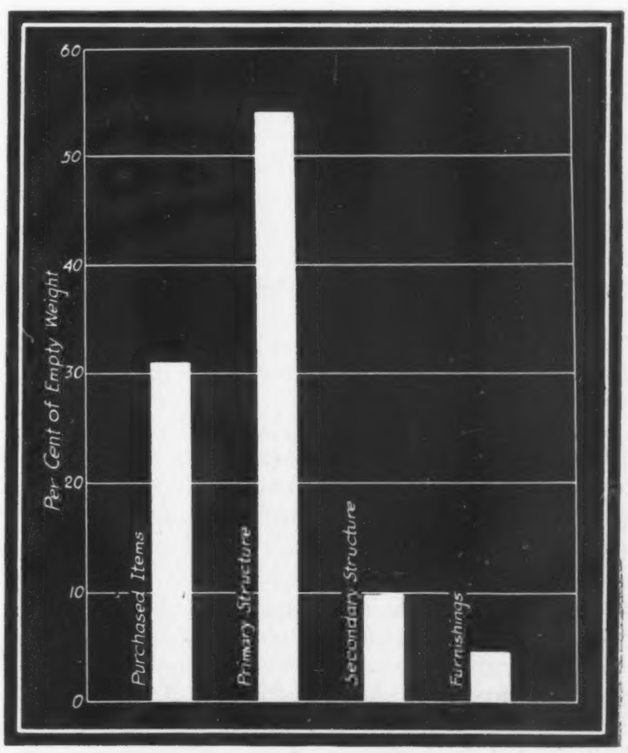


Fig. 3—Another puzzler confronting designers of aircraft: How to reduce weight, yet maintain strength and top performance? Here's weight breakdown of an Aeronca model

sufficient practice, but since this condition is somewhat similar to a beginner learning scales on the piano before he can play music, it is rather boring. Considerable progress toward the answer to this problem was made immediately prior to the war in a ship which had the two controls interconnected so that the airplane automatically made the correct turns. A modified form of this system is sure to appear on some postwar private aircraft since the airplane is then much easier to fly.

Accessories, such as flaps, slots, and spoilers, will be used where occasion demands, and will be automatic in operation wherever possible to give the pilot the safety advantage each one offers and yet not burden him with a complicity of levers to operate while in flight.

Difficulties in Reducing Price

Although it is easy to put down list prices for postwar aircraft, as I have already done, fulfillment of the goal set is not in keeping with our prewar experience. When it is considered that the prices we are aiming at are only slightly higher than those paid for automobiles, and that aircraft dealers are entitled to as good a profit margin as the automobile dealer, it becomes apparent that these prices are going to be very hard to meet with the small production of the aviation industry as compared with that of the automobile industry. However, a careful study leads us to believe these prices are not out of line.

As shown in *Fig.* 2, approximately 55.2 per cent of the airplane manufacturer's cost is beyond his control in the form of purchased equipment and material. Since this percentage is more than half the total amount allowed the manufacturer, he has two strikes against him at the start unless he can obtain the cooperation of his parts and materials suppliers.

Cost Control and Performance

Our postwar aircraft will be designed with initial cost controlled as rigidly as other factors have been in the past. Further, aircraft models will be carried over from year to year, with minor changes, to amortize the high development and tooling costs which will be necessary in order to start out with a reasonable price.

Spectacular increases in performance over prewar models will be had only at considerable additional cost. But honest cruising speeds of 100 to 120 miles per hour in the lower priced models will be a reality, with perhaps 120 to 140 miles per hour cruising speeds for the higher priced models. Landing and take-off speeds of 40 to 50 miles per hour will be necessary to allow the higher cruising speeds, unless considerable additional cost in the form of complicated high-lift devices and additional power is satisfactory.

The performance of an airplane depends a great deal

(Concluded on Page 172)

Current Availability of Metals and Plastics

IN THE accompanying table the current relative availability of the important metals and plastics essential to the war program is indicated. Taken from the War Production Board's quarterly Material Substitutions and Supply List No. 11, the tabulation shows three degrees of criticalness. Supply of those materials in Group I is *insufficient* to satisfy war plus essential industrial demands. Those in Group II are currently *sufficient* to satisfy these same needs. Group III materials are *in excess* of current *essential* needs.

Group I (most critical)

Metals, alloys, etc.

- Bismuth
- Cadmium
- Platinum
- Sodium
- Tantalum
- Tin
- Columbium
- Nickel
- Malleable cast iron

Synthetic Resins and Plastics

- Acrylic resins
- Alkyd resins
- Cellulose acetate
- Cellulose acetobutyrate
- Ethyl cellulose
- Phenolic resins
- Polystyrene
- Vinyl resins

Group II

Metals, alloys, etc.

- Aluminum
- Beryllium
- Copper
- Lead
- Magnesium
- Silver
- Zinc
- Steel; except items in Group III

Synthetic Resins and Plastics

- Melamine:
  - Aldehyde plastics
- Urea:
  - Aldehyde resins
  - Formaldehyde molding materials

Group III (least critical)

Metals, alloys, etc.

- Antimony
- Calcium
- Gold
- Iron:
  - Cast
  - Pig
- Mercury
- Palladium
- Cobalt
- Ferroboron
- Ferrochromium
- Ferromanganese
- Ferrosilicon
- Ferrotitanium
- Ferrovanadium
- Molybdenum
- Silicomanganese
- Silver iron
- Zirconium ferro-alloys
- Steel rerolled rail
- Steel reinforcing

Synthetic Resins and Plastics

- Cellulose nitrate
- Lignin

Because of limited manpower or processing facilities, certain metal fabricated forms are less free than the metals themselves. Currently such products would include: Cold-drawn seamless steel tubes, steel plates, wire rope, rails, hot-rolled sheets No. 18 gage and thicker; malleable iron castings; fine insulated copper wire, large-size copper tubing; and aluminum permanent-mold castings. The shortages in manpower require that greater emphasis be placed on the use of materials best suited to low unit labor processes such as stamping, die molding and die casting, production on automatic screw machines, etc.

When large metal tonnages are involved, first consideration still should be given to the ferrous materials to avoid disrupting the supply of smaller tonnage materials.

Compared with previous listings, the most significant changes are the easing of copper and steel to Group II. On the other hand most plastics have become slightly tighter. An exception to the general easing in nonferrous materials is tin, because of its continued critical supply. Use of tin in bronzes and plating should therefore be avoided wherever possible.

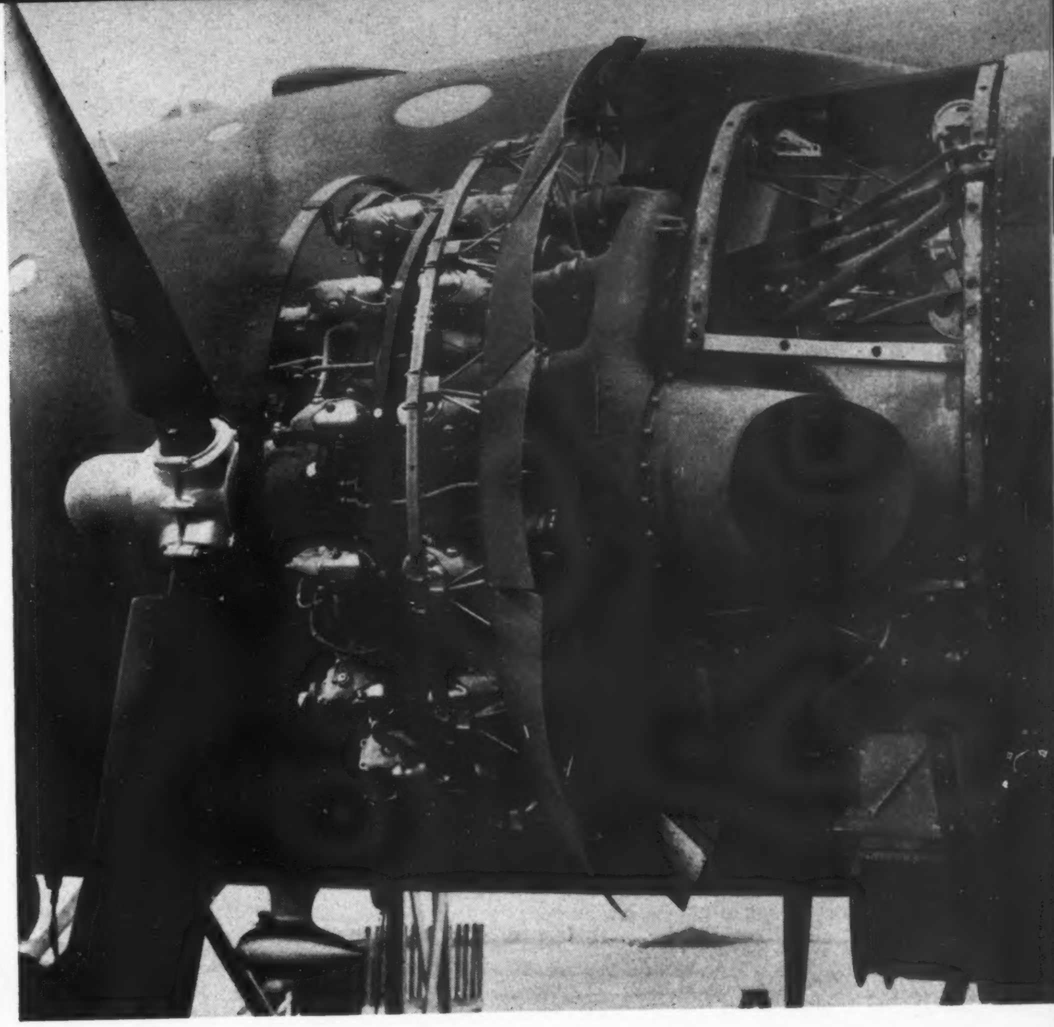


Fig. 1 — Right — Universal-joint manifold on airplane

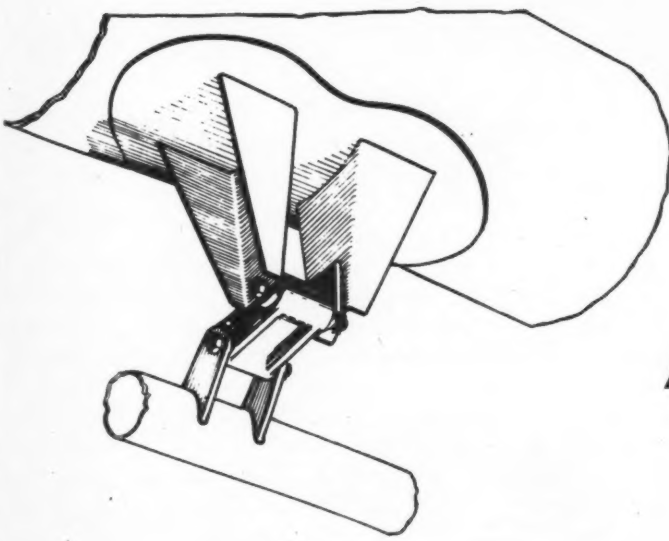


Fig. 2—Below—Typical ball and socket collector mounting using link attachment

Improving Aircraft Manifold Design

By Ralph L. Haver
Project Engineer
Exhaust Manifold Division
Ryan Aeronautical Co.

EFFORTS of aeronautical engineers to design airplanes with greater performance has led to tremendous improvements in power plants. Accompanying these advances and the resulting invasion of high altitudes have come new uses for exhaust gases. Wing anti-icing, gun installation heating, warming of surface controls and turbosupercharging have put to good use a source of energy which formerly was wasted.

These developments together with the overcoming of problems of vibration and expansion, to mention just two resulting from the use of higher horsepower engines, make the exhaust system of today a complex and exacting one. The methods developed for handling aircraft exhausts, no doubt, will find many applications for handling gases in process machines.

To develop an exhaust system that is efficient and which has the best maintenance characteristics combined with long service

life, design must be undertaken at the time when initial layouts are made and consideration is first given to the power plant installation as a whole. This will tend to eliminate the necessity of designing the exhaust system around every other major part which would result in loss of efficiency, excessive maintenance requirements and additional weight; it also might affect the operating efficiency of other major parts of the power plant assembly.

At the present time there are three general types of exhaust systems. First is the universal-joint type of manifold, commonly referred to as ball and socket manifold, which is composed of two or three sections bolted rigidly together and mounted on the engine mount ring or inner cowl by a series of links. The manifold is connected to

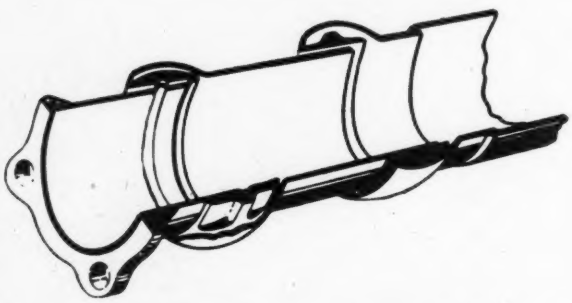
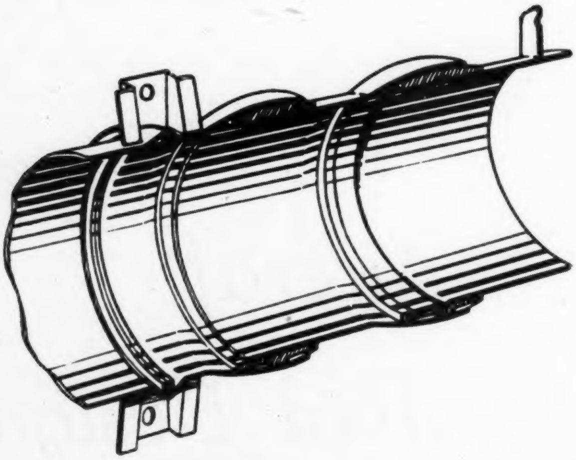


Fig. 3—Above—Cutaway view of patented ball and socket joints for a manifold

Fig. 4—Below—Cross section of joint employing detachable rear socket



the engine through the universal joints which absorb engine movement, vibration and expansion. An installation of this type is shown in *Fig.* 1.

Second type is the conventional slip joint manifold composed of individual sections mounted on the engine exhaust port and connected by means of collars which act as expansion joints. Third, there are the short stacks or injector stacks. This system, as the name implies, consists of individual stacks for each cylinder or pair of cylinders and is at present used mainly on military aircraft for increased performance and flame dampening.

Introduction of the decoupled type of vibration isolation suspension for high horsepower engines brought about the need for an exhaust manifold system which would perform with low maintenance requirements. Such a manifold, to be satisfactory, must provide adequately for the handling of inherent movement of the suspension as well as the expansion and contraction of the system.

The patented ball and socket joints used in the Ryan manifold are precision parts manufactured to close tolerances to maintain a leakage efficiency of approximately 95 per cent at a differential pressure of 30 inches of mercury. Essentially the ball and socket collector is a complete ring with no expansion or slip joints. Joints are provided only for assembly and are rigid mechanical connections. The collector body is mounted on the engine mount ring or inner cowl by a series of links, *Fig.* 2.

Typical ball and socket joints are shown in *Figs.* 3 and 4. A deflection of 7 degrees is the maximum recommended for a joint to accommodate the roll, pitch and yaw in its worst combination. Deflection has been increased to as high as 10 degrees but is not desirable.

Diameters of the collector ring body are determined by allowing .04 square inches of collector cross-sectional area for each horsepower from which exhaust gas is collected. For exhaust turbine installation this figure may be cut to .035 square inches per horsepower, and in some cases .03 square inches is allowable, providing the complete collector design is carefully executed to avoid excessive increases in back pressure at individual cylinders. The radius of the collector ring's center line is made as

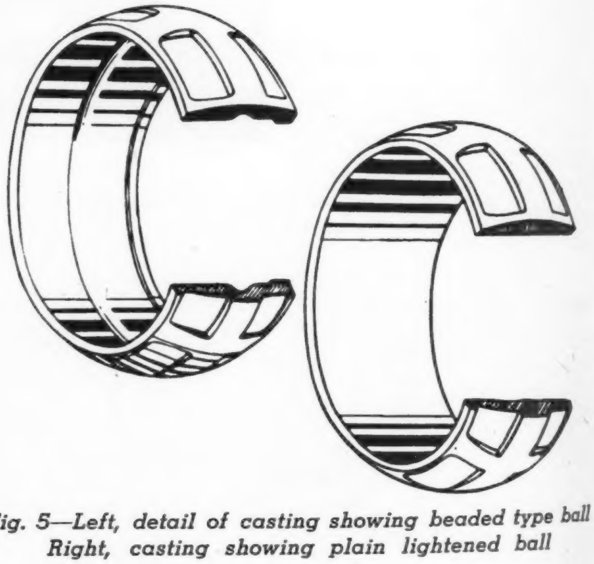


Fig. 5—Left, detail of casting showing beaded type ball Right, casting showing plain lightened ball

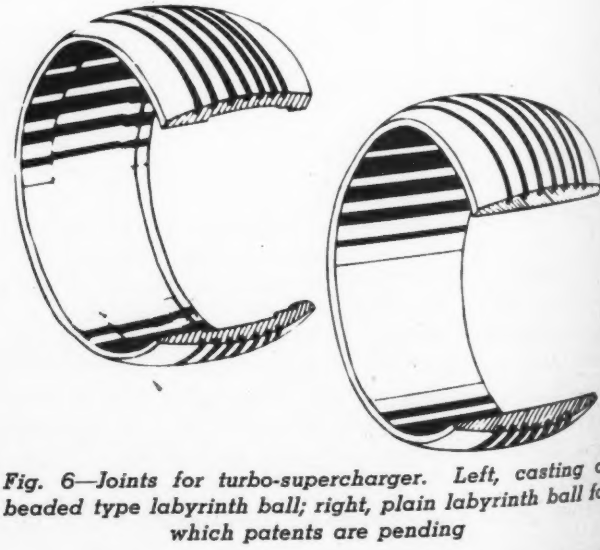
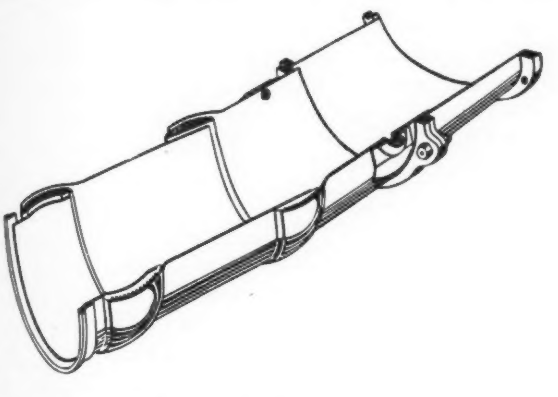


Fig. 6—Joints for turbo-supercharger. Left, casting of beaded type labyrinth ball; right, plain labyrinth ball for which patents are pending

near to the exhaust port radius as possible to reduce the complexity of manufacture.

Location of the outlet, tail pipe or exhaust turbine connection is determined by such items as the landing gear, wing location with respect to the nacelle, exhaust turbine, cowl flap design, oil coolers, and the nacelle structure, etc. This location does not materially affect the collector ring except that sufficient room should be allowed to flare the body of the collector into the outlet in a smooth curve. This is necessary to avoid hot spots and



to keep the turbulence of exhaust gases at a minimum and thus prevent back pressure resulting from excessive turbulence.

The cross-sectional area of the tail pipe is in general the same as that of the outlet or Y-section of the collector ring except where a special outlet is required on a tail pipe for flame dampening or where the exhaust gases are venturied into the exhaust turbine. Contour of the center line of the tail pipe or exhaust turbine connection is determined by the location desired for the exit of the exhaust gases, the exhaust turbine or other major part of the power plant installation and the nacelle structure itself. The tail pipe or exhaust turbine connection is kept as straight as possible. Elliptical sections are not used due to unsatisfactory service life. Care is also taken to keep the discharge point of exhaust gases in a low-pressure area to reduce the possibility of burning the surrounding structure.

Ball and socket arrangements eliminate hanging the collector on the engine exhaust port and permit the power plant to vibrate and oscillate free of the collector ring body since universal joints absorb the movement between the manifold and the engine. This type of installation adequately handles movement of engines using dynafocal or similar mountings.

Provisions must be made for mounting the ball and socket collector ring from the engine mount or inner cowl, preferably from the engine mount ring. This is done by providing brackets on the engine mount ring located in radial planes from which the collector is supported by a short link, *Fig.* 2. For most installations 1 to 1½-inch links have been found to be satisfactory, these being used mainly for standardization purposes. Six or seven links are sufficient for mounting this style of collector and should be located systematically around the collector where possible. Bushings are desirable in the links to prevent binding in the hanger brackets. With this type of manifold the engine exhaust ports support approximately 15 pounds of the collector ring weight, distributed over the full 9, 14 or 18 cylinders as the case may be. Remainder of the weight is supported by the engine mount or inner cowl.

The universal joints are essentially a tube which is spherical on one end, enclosing a Ni-resist spherical sleeve or ball. They may be solid rings lightened by means of depressions located radially for light-duty collector ring installations, *Fig.* 5, or solid rings lightened by annular grooves on the outside surfaces. The annular grooves or labyrinths are specially designed to give maximum sealing action against leakage of exhaust gases, *Fig.* 6. This latter type is used for exhaust turbine installations or other installations requiring minimum leakage at high differential pressures. Support tube or exhaust nipple in turn fits inside the ball, making the joint complete. Because this design does not require collars to connect each port section and utilizes lighter gage material for the body, weight of the manifold is on a par with that of the slip-joint type.

Installation is simplified because the manifold body can

Fig. 7 — Above — Cut away of joint used between collector outlet and turbo-supercharger

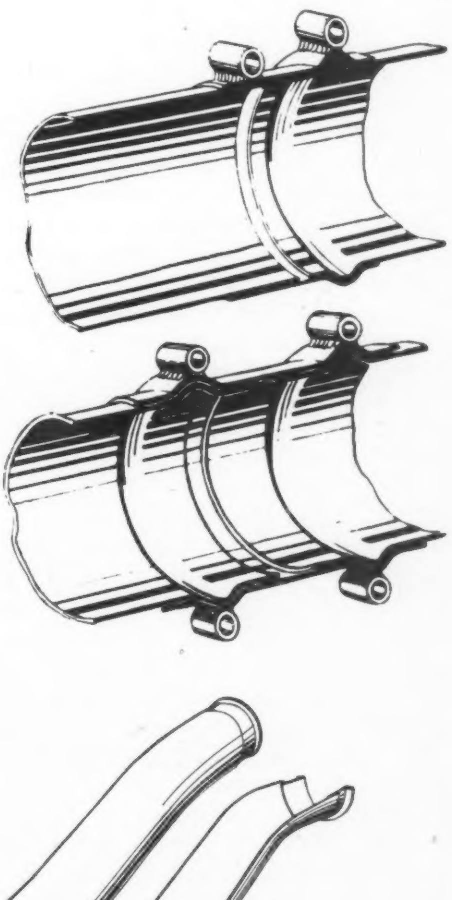


Fig. 8—Right—Single-beaded connection and double-beaded connection for slip joints

Fig. 9—Below—Cantilever type jet stack propulsion

be installed on the engine mount or cowl well before the engine is swung into place. After installation of the engine the port tubes are bolted into place and the collector ring adjusted and secured. This permits the installation of cowling, tail pipes, exhaust turbosupercharger connec-

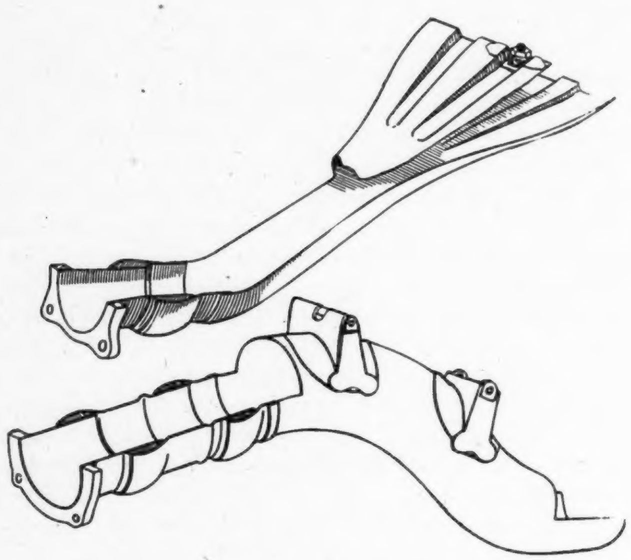


Fig. 10—Cutaway of patented single and double-ball ball joint short stack

tion and other items prior to actual engine installation.

Expansion joints, slip joints and complicated vibration joints between collector and tail pipes are eliminated. There is no vibration to be allowed for in the connections to a tail pipe or turbine. Only linear and radial expansion must be considered. This is particularly advantageous on exhaust turbine installations since vibration, which would be detrimental to the turbine's life and operation, is not transmitted to it. The expansion of the exhaust collector and the exhaust turbine can be taken care of through the use of large ball and socket joints placed between the two units, *Fig.* 7.

At connections between the turbosupercharger and the collector ring in manifolds other than ball and socket types, complicated vibration joints must be utilized to protect the exhaust turbine. These joints usually are packed with braided metallic packing which requires frequent replacement because of the compacting action of imposed vibration. Ball and socket collectors, however, do not require this type of packed joint to obtain effective sealing action because of the absence of vibration in the collector ring body which is decoupled from the engine through the medium of ball and socket universal joints. Due to the fact that strains and stresses in a slip joint—caused by engine vibration and deflection—are not present in the ball and socket collector, the life of the manifold body is increased and the need for heavy-gage material is greatly reduced. In the case of exhaust turbine installation .043 material is all that is required as compared to the .05 gage needed for the slip-joint collector.

A manifold made from .031 stainless steel was installed on an experimental engine for test purposes and was run intermittently for over 500 hours. Inspection of the manifold showed it to be in good condition and apparently capable of indefinitely prolonged service.

Columbium stabilized 18-10 stainless steel is used on these ball and socket manifolds. Use of Ni-resist in connection with the stainless offers an ideal combination in that no galling of the adjacent metal occurs. In dismantling the engine, the port nipple is detached from the engine and pushed back, thus telescoping the ball and socket joint and permitting the removal of the cylinder. The mechanical joint at the rear socket allows the removal of the entire joint if desired.

Design of slip-joint collectors differs in some respects from ball and socket types. The slip-joint collector is located near the engine exhaust-port face to reduce the overhang of the collector body and resultant stresses from engine movement and vibration. The collector ring must be broken into individual sections, one for each cylinder or each two cylinders depending on the type of engine used in the installation with a slip-joint connection, *Fig.* 8, inserted at each break.

Clearance provisions must be made to allow for movement of the collector ring body and outlet with respect to other parts of the installation since it is attached direct to the engine. Where long tail pipes or exhaust turbine connections are used, this movement and vibration must be isolated from the tail pipe or exhaust turbine. Provision must also be made for differential expansion in the exhaust port legs for twin-row installations. This is usually done by inserting a slip joint or expansion joint in the front cylinder port leg, *Fig.* 8. In some cases, to assist in assembly and economize on material, the rear port leg is also joined to the collector body section by a double-beaded collar, also shown in *Fig.* 8.

Short Exhaust Stacks

The short stack or ejector stack type of exhaust systems has opened up an entirely new field of design. While this type is relatively new, much experience and test data have already been accumulated. At the present time each installation is a special design problem. Outlet, shape of the individual stack and method of attachment are all determined by the engine cowl flap design, inner cowl design, location required for the outlet due to the wing location, landing gear doors, etc. Data on the areas required for each type are fairly well established and can be relied upon to be accurate.

In general, short ejector stacks may be mounted direct on the engine exhaust ports, *Fig.* 9, providing the flap design and gill opening is such as to allow for the movement of the stacks with the engine or provisions are made for cutouts in the engine cowl. Long ejector stacks should be mounted partially or wholly on the nacelle structure, depending on whether a single or a double, *Fig.* 10, universal joint is employed. This is necessary due to the high stresses introduced and failures resulting from engine movement and vibration. "Siamezing" of ejector stacks is not recommended. In some cases where it is absolutely necessary due to lack of room it may be done, but care should be taken to include only those ports that fire at least 180 degrees apart and if possible 360 degrees.

The ever-increased demand for flame dampening exhaust equipment essential for night combat flying is already being met to a certain degree and will no doubt be fully met in the immediate future.

A New Slant on Sheet-Metal Fastenings

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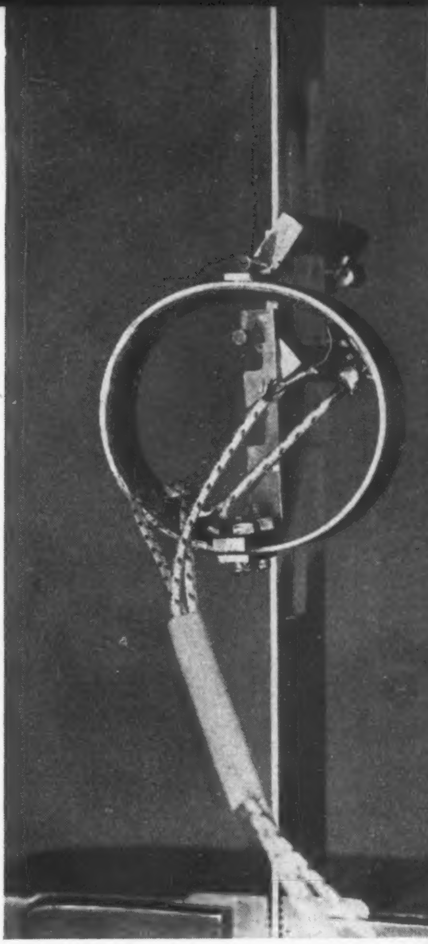
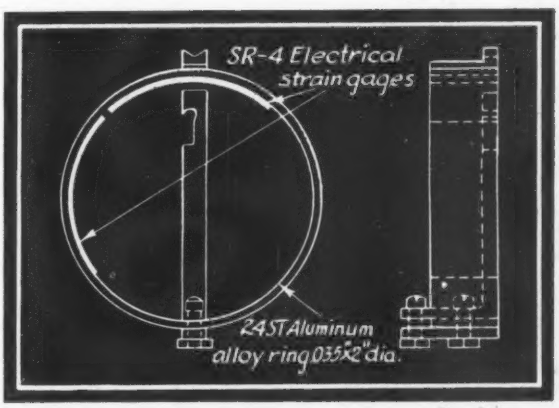


Fig. 1—Above—Special gage is shown mounted on a spotweld specimen for recording extension between two spotwelds

Fig. 2—Below—Details of ring gage for measuring one-half inch gage length deformation



IN THE design of joints and fastenings, the engineer has adopted a strictly pragmatic philosophy. For joints loaded concentrically, he uses the rule that the load is evenly distributed between the fasteners; for those eccentrically loaded, rules of the same order of simplicity have been formulated. Despite the fact that their bases are largely empirical these design methods have the sanction of almost universal usage. Whatever success they achieve is due largely to a knowledge of strength test data, interpreted according to the rules used in design. Recently, it has become generally recognized that extensive extrapolation of the methods often lead to erroneous conclusions, since they do not constitute true solutions to the structural problems involved in the design of joints.

Illustrative of the complexity of the problem are the results of a test program recently conducted in the laboratories of a leading aircraft manufacturer. The object was to determine the strength of a row of removable fasteners, placed parallel to the direction of the tensile load. The somewhat disconcerting test data indicated that the use of an increased number of fasteners did not result in a corresponding increase in the strength of the connection.

Existing Rules Contradicted

These results contradict one of the most universally used rules of riveted design, namely, that each rivet of a noneccentric pattern carries an equal portion of the total load and that the strength of each rivet in the group can be obtained from tests of one or more rivets. While this rule was originated in connection with riveted joints only, its simplicity and clarity strongly suggested the possibility of its extension to cover other types of fasteners as well. Such an extension, in the case of the removable fasteners, is invalid. It is invalid for practically all fasteners since it oversimplifies the analysis of the joint. The engineer should realize that the design of fastened joints involves the study of a redundant structural system.

For ease of mathematical manipulation, an analysis of the structural behavior of fastenings is best divided into at least two parts. The first is a study of continuous connections, such as cemented joints; the second deals with discontinuous connections, as exemplified by multi-rivet and multi-spotweld joints. The former utilize the entire contact area between the two parts for bonding, as compared with the partial bonding which exists in the latter. The two naturally are interdependent to some extent.

Considering discontinuous connections, a discussion concerned specifically with riveted joints gives some initial insight to the problem. It is of particular interest to discuss the load distribution in multi-rivet joints and to ex-

From a paper presented at the recent War Engineering annual meeting of the Society of Automotive Engineers in Detroit.

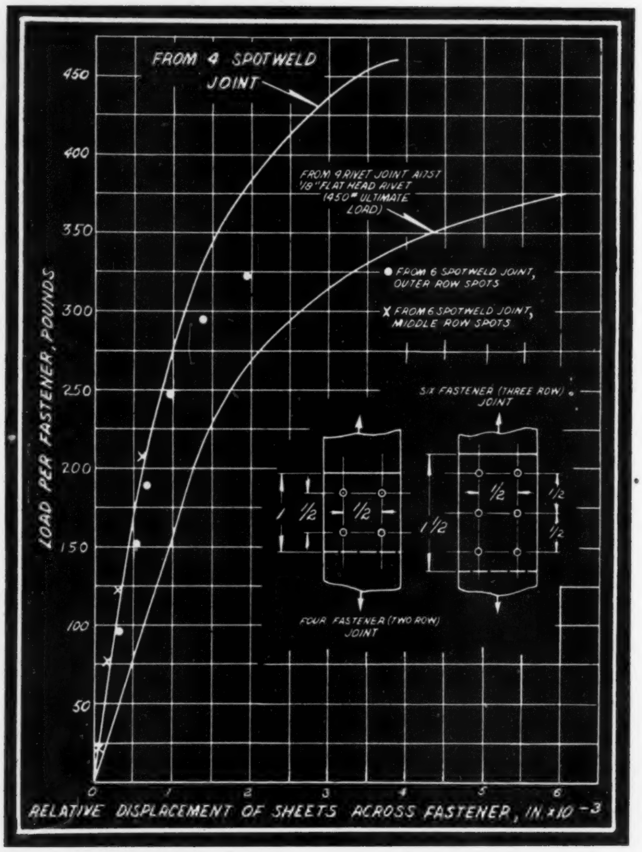


Fig. 3—Load-deflection curves for spotwelds and for a rivet. Spotwelds were made on Alclad 24-ST sheet

plain how load equalization on all rivets is brought about at high joint loads.

It is readily recognized that large, compressive bearing stresses exist in the frontal region of a loaded rivet. Effect of these high stresses is to cause early yielding of the sheet material surrounding the rivet, with consequent distortion of the circular rivet hole to a somewhat elliptic shape. If the deformation across a single rivet of one of the joined sheets relative to the other is measured (such a deformation is shown as δ_1 on *Fig.* 11), the yielding of the sheet around the rivet is evidenced by a reduction in the slope of the load-relative deflection curve; given load increments are accompanied by deformation of increasing magnitude as larger portions of the sheet become plastic. Apparent "stiffness" of the rivet decreases.

Load Distribution Affected by Stiffness

Consider now two rivets, in line with the load and unequally stressed. Since the more highly loaded rivet has the least stiffness, it will pick up load at the slowest rate. The greater stiffness of the less highly loaded rivet causes it to pick up load more rapidly. This inequality of load gains will clearly persist until a joint load is reached at which both rivets achieve equal loading, or at which premature fracture occurs. The facility with which load redistribution takes place is found to be dependent on the slope of the load-relative deflection curve for the fastener and on the stiffness of the sheets between fasteners (defined as Ewt/l, where E = Young's modulus for the sheet material, w = transverse fastener spacing, t = sheet thickness and l = longitudinal fastener spacing in direction of load). A large sheet stiffness and a marked yielding tendency of the fasteners leads to the most rapid equalization of load. These conclusions are based on the development given in the Appendix, where the problem of the load distribution in a multi-fastener joint is treated in detail.

Since rivets do exhibit marked yielding tendencies at high load, each rivet will carry an approximately equal share of the joint load at fracture. Initial imperfections in the joint, such as an improperly upset rivet, will affect the load distribution only during the low load range; as soon as the initial clearance is taken up by plastic slippage of the remaining rivets, the unloaded rivet will rapidly pick up its share of the load. The fracture load distribution will be similarly unaffected by elastic stress concentrations which are introduced purely by virtue of the rivet pattern employed; they will be damped out in the high load range by the ductility of the fasteners.

Large bearing stresses, which account for the ductility

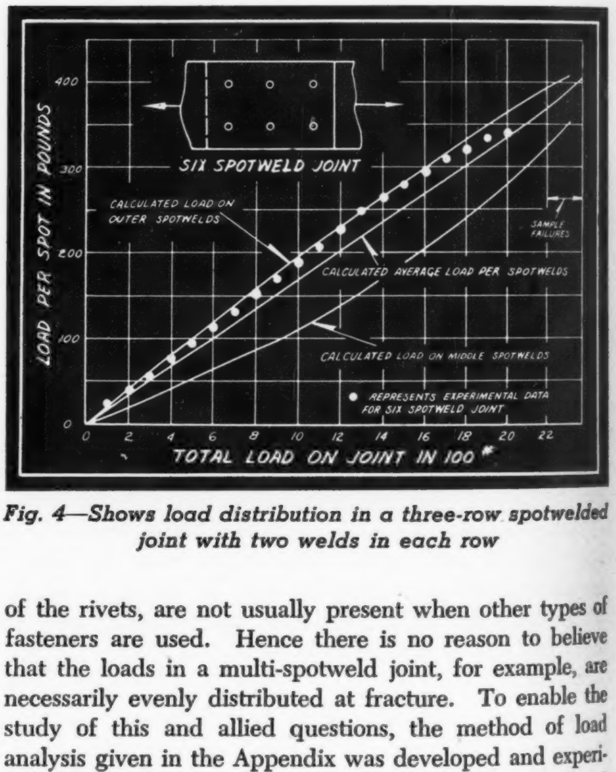


Fig. 4—Shows load distribution in a three-row spotwelded joint with two welds in each row

of the rivets, are not usually present when other types of fasteners are used. Hence there is no reason to believe that the loads in a multi-spotweld joint, for example, are necessarily evenly distributed at fracture. To enable the study of this and allied questions, the method of load analysis given in the Appendix was developed and experimentally checked. It is based on several easily visualized statements concerning the relative deformations across the fasteners. Since the method is based on equalities of deformations, which are directly measurable quantities, it is well adapted to experimental check.

Object is to enable rapid and accurate estimation of the load distributions at the yield and fracture of a multi-fastener joint. Accordingly, emphasis has been placed on the development of a simple rule which can readily be used in practical design. The present method satisfies this requirement. Furthermore, it presumes the knowledge of no empirical data save the load-relative deflection curve, obtained by test, for an individual fastener of the type under consideration.

Methods similar to those outlined have been suggested in the past by numerous investigators. One of the major

difficulties encountered in their application was the accurate determination of load-relative deformation data for the fasteners, especially in the plastic range just preceding fracture. Instrumentation problems are great; extremely small deflections must be accurately measured, using gages which preferably are not destroyed by fracture of the specimen. No existing gage of requisite accuracy and design, capable of withstanding fracture, was known to us at the start of the test program.

Measuring the Deformations

Consequently, a new ring-type gage was constructed which can be suitably modified to record overall deformations of the joint, deformations of the sheet between fasteners or load-relative deflection data across individual fasteners. *Fig.* 1 shows one of the gages mounted on a spotweld specimen, arranged to record the sheet extension between two spotwelds. *Fig.* 2 is a drawing of the gage, illustrating the details of its construction. To use the gage, two lugs are first clamped to the specimen at the gage points, as shown in *Fig.* 1, the clamps having tempered steel points. The ring gage is then positioned between the two lugs in a state of initial compression. As the specimen is loaded, the lugs move apart and relieve

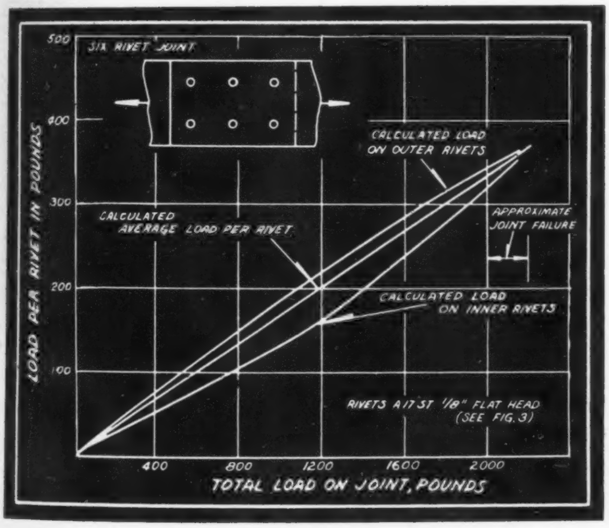


Fig. 5—Above—Load distribution for a three-row riveetd joint shows greater uniformity than for a similar joint with spotwelds, due to decreased stiffness

Fig. 6—Below—Dimensions and stresses for a cemented joint. Actually, length l would be several times 2c

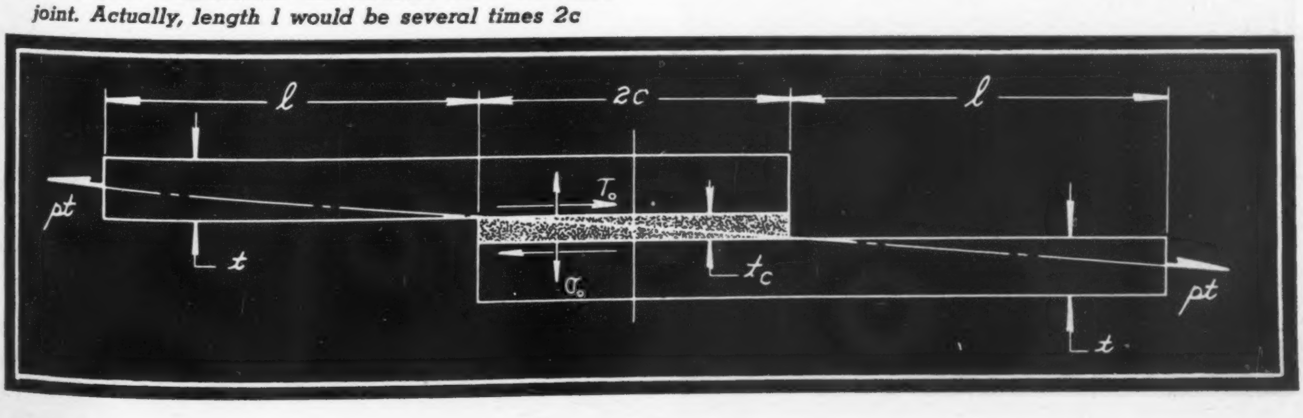
the compression of the ring. This in turn activates two electric resistance type strain gages cemented to the ring surface (*Fig.* 2). The strain gage indications are read with standard equipment and are interpretable directly in terms of specimen deformations. Since the outer ends of the gage grooves, in which the lugs are positioned, are built with large clearance, fracture of the specimen in a controlled-strain type testing machine does not harm the gages in any way. It is to be noted that the gages are positive acting throughout their range. The multiplying error of the gage and lug system used in the tests about to be described has been estimated as approximately 5 per cent. A large portion of this error, it is to be noted, is due to the lug attachments; the gage alone possesses far superior accuracy.

Shown in *Fig.* 3 are the load-relative deflection curves obtained across individual spotwelds located in two and three-row joints. Each curve is the average of many tests and the agreement between the curves for different samples is within the limits of the gage accuracy. It is to be noted that the load-deflection curves for all three spotwelds are clearly identical. The early bending over of the curve for the outside spotwelds of the three-row joint is explained by the excessive yielding of the entire sheet outside the joint as the failure load is approached. This condition does not occur in the two-row joint until much higher spot loads are reached. Agreement of the earlier parts of the curves fairly conclusively shows that the behavior of an individual spotweld is not dependent on the joint pattern, although premature yielding of the outside spots may sometimes be expected. Proof of this fact is essential if the methods of analysis employed in the Appendix are to be used successfully.

Rivet and Spotweld Stiffness Compared

Also shown on *Fig.* 3 is a load-deflection curve for a rivet with an ultimate strength equal to those of the spotwelds used in this study. The greatly decreased stiffness of the riveted connections compared with the spotweld is apparent. Deformation data for the rivet were taken to the ultimate load, although not shown on *Fig.* 3 because of the large magnitude of the range of abscissa (deformation) values which would be required.

It should be noted that data for one-row joints are not given. The excessive joint bending which occurs when one-row connections are loaded tends to make the action of this joint markedly different from those of multi-row joints. Accordingly the load-relative deflection data obtained for the spotwelds in two-row joints are taken as



representative of the action of all individual welds in multi-row joints; the one-row data are not acceptable for this purpose.

Agreement between the calculated and measured load distributions in spotwelded three-row joints is shown in *Fig.* 4. The calculated distributions are based on the load-deflection curve for an individual spot obtained from two-row joints and on the methods of the Appendix; the manner of computing the experimental points also is outlined in the Appendix. Agreement between the two is excellent, save in the load range just preceding fracture where the measured values show a greater redistribution of load (toward uniformity) than do the calculated. At a joint load of 45 per cent of the ultimate load, for example, the measured and calculated load distributions are in almost perfect agreement and show that:

a. The outer row spots are 71 per cent more highly loaded than are those in the middle row

b. The outer row spots are loaded 16 per cent more highly than if the load were uniformly distributed.

c. The middle row spots are loaded 32 per cent less highly than if the load were uniformly distributed.

At a joint load equal to two-thirds of the ultimate, corresponding to the arbitrarily defined "yield" load in aircraft structural analysis, the inequality of the loads carried by the spotwelds is equally apparent. The outer spots are 53 per cent more highly loaded than are the middle spots; the outer row spots are loaded 13 per cent more highly, and the middle spots 26 per cent less highly than they would be if the load were uniformly distributed among the fasteners. If the fastener ductilities were increased so as to cause the middle row spot to be as highly loaded as is the outer spot at this load condition, *the joint strength would be increased by* 14 *per cent.* This last figure is of extreme importance, since it represents the optimum load advantage which can be gained by producing spotwelds of greater ductility than those employed in this program.

At the fracture load, both the experimental and measured values indicate that the load distribution among the spots is practically uniform. It is to be noted, however, that this is a relatively unimportant conclusion; the joint design is based on a joint load which is a fraction of the ultimate load. At these lower loads, the effects of the load concentration in the outer spots cannot be ignored.

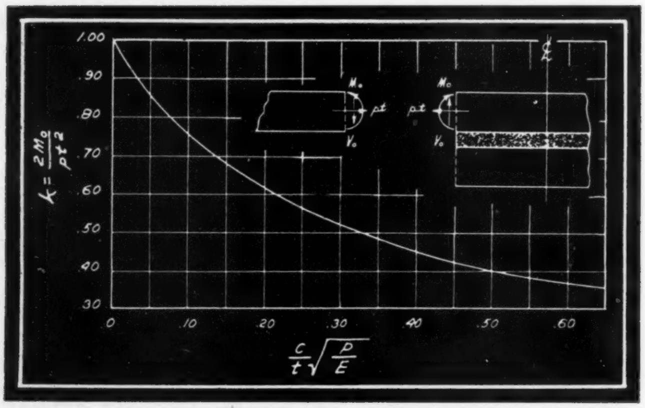
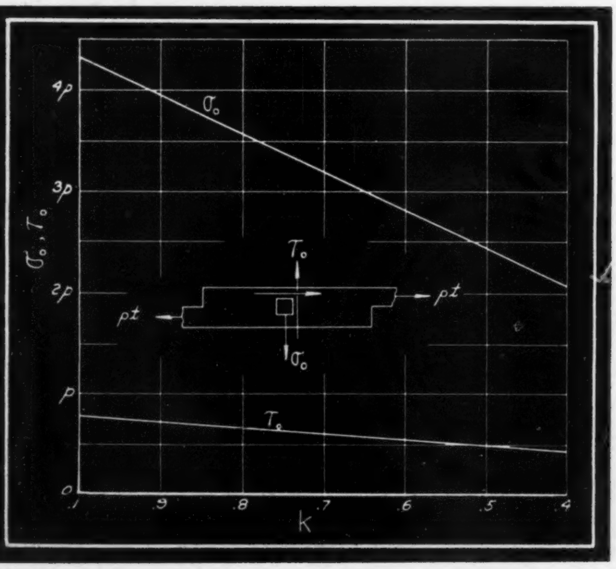


Fig. 7—Above—How the joint load factor k varies with joint proportions and load for a cemented fastening

Fig. 8—Below—Variation of normal and shearing stresses in the cement for a joint with a thin cement layer



Rivet Loads More Uniformly Distributed

Load distribution in a three-row riveted joint is shown in *Fig.* 5, based on the load-deflection curve for a rivet as shown on *Fig.* 3. It is clear that the decreased stiffness of the rivet, compared with that of the spotweld, causes the load in the joint to be more nearly uniformly distributed than in comparable spotwelded joints at all values of the total joint load. The load concentrations are, nevertheless, not entirely negligible. At two-thirds of the joint ultimate load, the outer rivets are loaded 28 per cent more highly, and the middle row rivets 15 per cent less highly than they would be if the load was uniformly distributed. Increasing the rivet ductility so as to cause the middle rivet to become as highly loaded as is the outer rivet at this load condition would cause an increase in joint strength of only 7 per cent. At fracture of the joint, *Fig.* 5 shows all of the rivets to be uniformly loaded, as would ordinarily be expected.

It can be concluded, therefore, that the design of spotwelded joints by assuming a uniform load distribution in the joint introduces errors of the order of 15 per cent; riveted joint design by similar methods introduces design errors of the order of only 5 per cent. Furthermore, the lesser stiffness of the riveted connection makes the load distribution in riveted joints more efficient than the corresponding distribution in spot-welded joints, for those which have been studied.

Continuous Joints Precisely Analyzed

For the continuous, cemented joint, we have found it possible to determine analytically the elastic stress distribution in the cement layer with great precision. The mathematical methods utilized in the solution, in keeping with the difficulty of the problem are extremely tedious and complex. Nevertheless, a number of interesting conclusions are drawn from the results and are significant with regard to both continuous and discontinuous joints.

The problem is formulated as shown in *Fig.* 6. Two sheets of thickness t are cemented together with an overlap length $2c$. The joint is considered to be wide in the direction perpendicular to the plan view of *Fig.* 6 and the sheets are assumed to extend a distance l to either side of the lapped region, the magnitude of l being at least several times the lap length of $2c$. Physical properties of the sheet material are denoted by E, G and ν, representing Young's modulus, shear modulus and Poisson's ratio, respectively. Both the sheet and the cement are assumed to be elastic. The cement layer thickness is called t_c and the physical properties of the cement are E_c and G_c.

Comparing Induced Stresses

The stresses in the cement which are found to be of importance are indicated schematically on *Fig.* 6. They are the shearing stress τ_0 in the plane of the layer, and the normal stress σ_0 in a direction normal to this plane. The normal stress in the cement, parallel to the cement plane, is found to be of relatively unimportant magnitude. When writing the values of τ_0 and σ_0, it is convenient to compare their magnitudes with that of the mean normal stress p which is applied as a tensile load in the two sheets.

The problem has been divided into two parts:

1. Where the cement layer is so thin that its effect on the flexibility of the joint may be neglected
2. Where the joint flexibility is mainly due to that of the cement layer.

The first solution is applicable to cases of cemented wooden and plastic sheets and is valid whenever:

$$\frac{t_c}{E_c} \le \frac{1}{10}\,\frac{t}{E} \quad \text{and} \quad \frac{t_c}{G_c} \le \frac{1}{10}\,\frac{t}{G}$$

The second case is significant for the analysis of cemented metal sheets and is valid whenever

$$\frac{t}{E} \le \frac{1}{10}\,\frac{t_c}{E_c} \quad \text{and} \quad \frac{t}{G} \le \frac{1}{10}\,\frac{t_c}{G_c}$$

The reasoning behind the establishment of these orders of magnitude relations, as well as the analytic details of the entire study, will be found in a paper, "The Stresses in Cemented Joints" by M. Goland and E. Reissner, presented at the 1943 Annual meeting of the Applied Mechanics Division of the American Society of Mechanical Engineers. The results given here are reproduced from this paper, which appears in the March, 1944 issue of the Journal of Applied Mechanics.

Before evaluating the stresses in the cement, it is necessary to calculate a "joint load factor", designated k. The value of k is a measure of the moment load transmitted by the two uncemented sheet portions to the lapped portion (referred to as the "joint"). If the factor k is multiplied by one-half the product of the applied tensile stress p and the square of the thickness of the sheets, the edge moments on the joint ends are obtained. In addition to these moments, it is readily seen that the applied tensile load will be carried through the sheets to the joint edges without change of magnitude. The

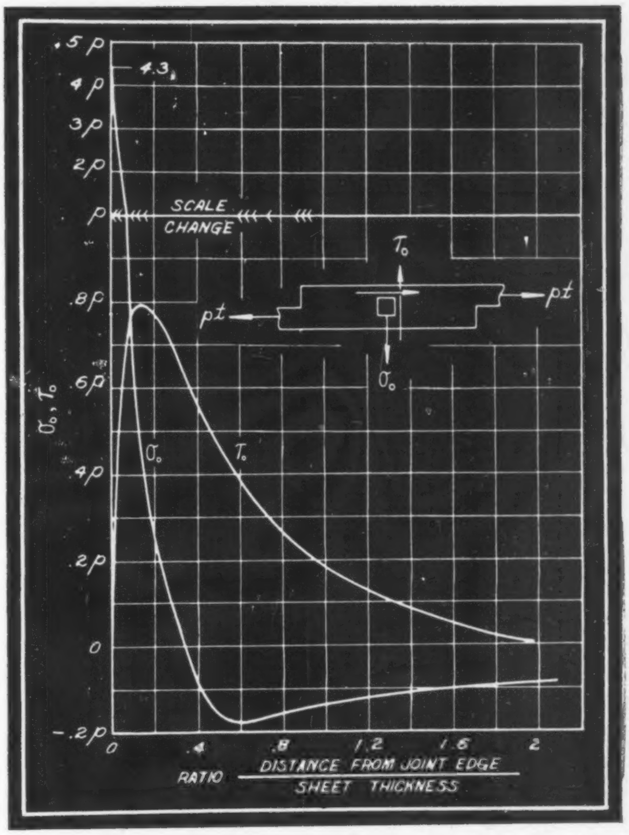
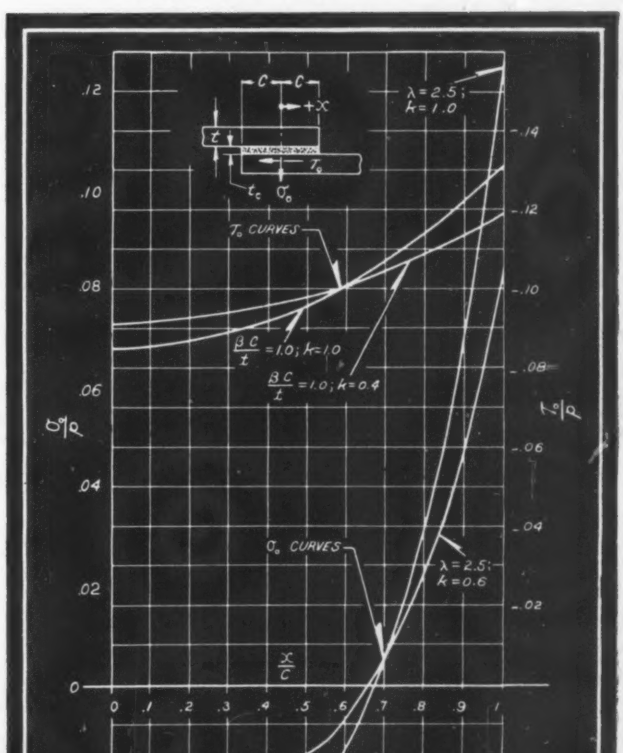


Fig. 9—Above—Shows stress distribution along the shear plane of joint with thin, inflexible cement layer for k=1

Fig. 10—Below—Stress distribution along the shear plane of a joint with a flexible sement layer



value of k is found to be dependent on the parameter $(c/t)(p/E)^{1/2}$, *Fig.* 7. Here c/t represents the ratio of one-half the joint overlap length to the sheet thickness. As applied joint load p increases, the magnitude of k is seen to decrease. The result of this is to cause the joint edge moment to increase with load, but at a reduced rate.

High Stresses Found in Cement

Maximum stress concentrations in an inflexible cement layer are shown in *Fig.* 8 plotted as a function of the factor k. Maximum cement stresses are surprisingly high, the σ_0 stress being from two to four times as large as p and the shear stress τ_0 varying from $.45p$ to $.79p$. The concentrations of stress are reduced as the joint load increases, that is, as k decreases, but the reduction of k is offset by the increase of p and is not sufficient to prevent the steady rise of the numerical magnitudes of the maximum stresses.

It is of extreme interest to note the existence of such large cement stresses, particularly with regard to the σ_0 stresses. During actual joint tests, it is observed, that the start of failure is characterized by a splitting apart of the two sheets at the joint edges. This action is explained by the presence of the high σ_0 stresses. For easy reference, a descriptive title for the σ_0 stresses is desirable; the name "tearing stresses" is suggested as being appropriate for this purpose.

Stress distribution along the shear plane of the inflexible layer joint in the vicinity of the edges is shown in *Fig.* 9, calculated for a value $k = 1$. The σ_0 stress is highest at the joint edge and rapidly damps out to a small value, $.1p$, within a distance along the shear plane of about two sheet thicknesses; the reversal of the σ_0 stress from a tensile to a compressive stress within this distance is of interest. The τ_0 stress has the value zero at the joint edges; it rapidly rises to a peak value and once more decays to almost zero within a distance of two sheet thicknesses. In joints with relatively inflexible cement layers, therefore, the tearing and shear stresses in the cement are concentrated in small regions near the ends of the joint. In the middle of the joint there is practically pure tension equal to $.5p$ in each sheet.

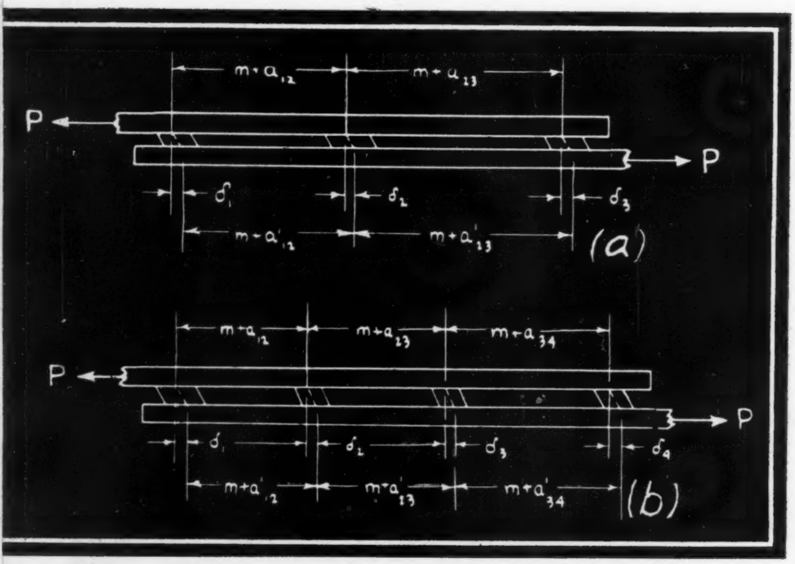


Fig. 11—Dimensions and deformations of three-row and four-row discontinuous joints

For joints with relatively flexible cement layers, results of entirely different nature are obtained. To begin with, whereas the stress distribution in the cement for the inflexible layer case is independent of both the joint dimensions and the physical properties of the cement material, the cement stresses in flexible layers are dependent on both of these factors. Furthermore, the cement stresses are no longer restricted to small end zones in flexible layers, but are of appreciable magnitude over the whole of the shear plane.

Fig. 10 shows typical σ_0 and τ_0 distributions in joint with flexible cement layers, calculated for several k values. The definitions of β and λ which appear on the figure are as follows:

$$\beta = \sqrt{8\frac{G_c}{E}\,\frac{t}{t_c}} \qquad \lambda = \frac{c}{t}\sqrt[4]{6\frac{E_c}{E}\,\frac{t}{t_c}}$$

The joints are composed of representative materials and dimensions (such as aluminum alloy sheets cemented with a typical resin adhesive and with a joint overlap to joint thickness ratio of five). It is evident that the increased flexibility of the cement layer causes the shear stress τ_0 in the cement to be more uniformly distributed over the joint shear plane. Nevertheless, the distribution still indicates a sizeable concentration of shear in the end regions of the joint. The stresses again reverse their sign near the joint edges, but are not damped away as quickly as they were in inflexible cement layers. The peak σ_0 and τ_0 values still persist at the joint edges.

Importance of Tearing Stresses

As previously indicated, the flexible cement layer analysis is applicable to joints composed of thin metal sheets cemented together. Adhesive materials used in such joints are particularly weak in tension, hence the σ_0 stresses, although smaller than those encountered in inflexible layer joints, are still of sufficient magnitude to cause a splitting failure at the joint edges.

Continuous joint results have been given in detail not only because of their interest with regard to cemented joints, but also because they provide a qualitative insight to the behavior of discontinuous joints. The significance of the presence of the large tearing stresses cannot be underestimated. They are undoubtedly at least partially responsible for the failure of the edge fasteners in spot-welded and other similar connections. They are also of tremendous importance in determining the fatigue characteristics of the connections. It is clear from the results that a careful consideration of the tearing stresses in every problem of fastener design is a necessary step toward its proper solution.

On the basis of the available results to date, we believe that a concrete start has been made toward the

(*Concluded on Page* 160)

Dreams of Today May Be Actualities Tomorrow!

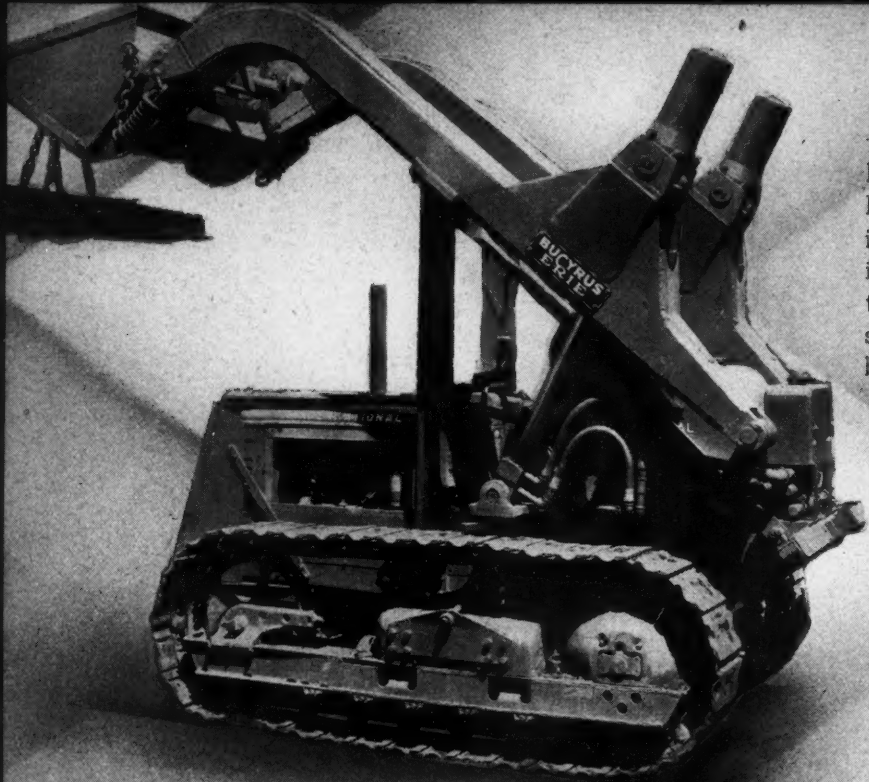
ELSEWHERE in this issue is published an illustration of a dream airplane of the future. The author of the article in which this is included—unlike many who have visualized postwar developments much too radically—makes no claim that the type of plane illustrated will mature in the early days following the war. He analyzes the possibilities of the future from the standpoint of current events and trends.

Such appraisals of potential developments and markets are being carried out more and more by machinery builders and are being used as the bases for long-term or short-term postwar programs. Many of these necessarily contemplate a complete changeover from the line of machines manufactured before or during the war while others envision a return to former standard lines as military requirements are eased.

One company in particular—to cite a case in point—has no hesitation in saying that, to take care of all contingencies as far as is feasible, it has set up as many as three separate and distinct postwar programs. The first of these, because of the necessity of returning to normal production as quickly as possible on reconversion, is based on prewar models. The second program involves design and production of improved models of the company's standard line, and the third visualizes the development of an entirely new line of machines not previously available and therefore capable of creating new fields of employment and wholly new markets. The second program, incidentally, is double-barreled in that it is aimed to meet either a low or high-priced economy as it may develop after the war.

Some "dreaming" necessarily is involved in all such postwar programs. There is no question that this should be indulged in to the extent that war requirements permit. Particularly is this the case insofar as design work is concerned, inasmuch as it is primarily upon design developments that the future success of machine-building companies is based.

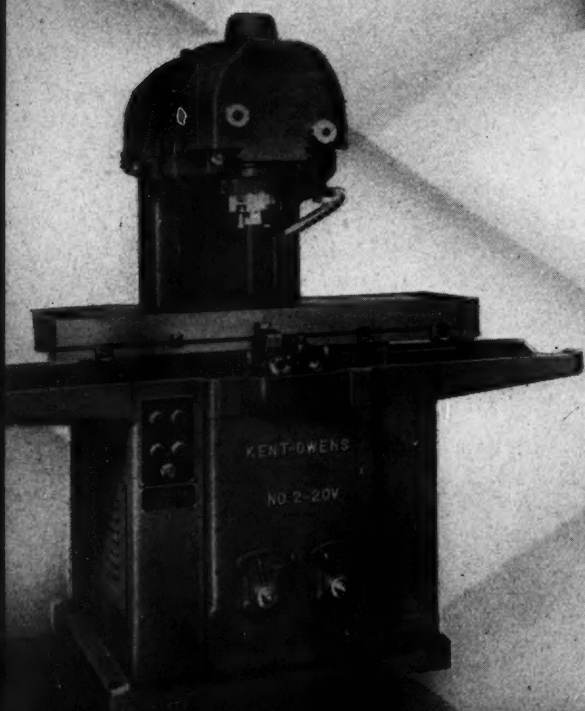
L. E. Jermy

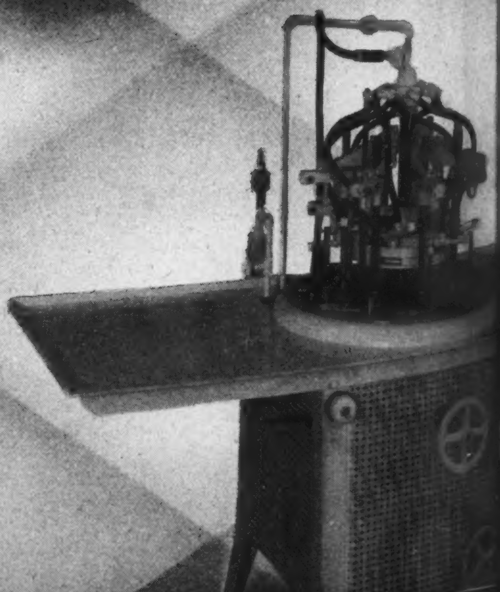


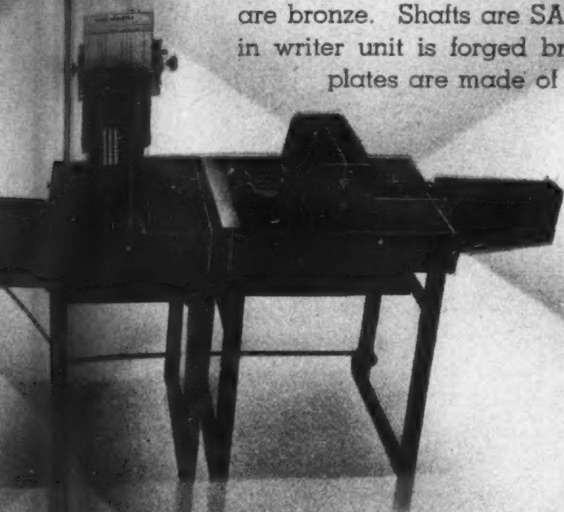
Left—With gear type pump, hydraulic lift of Bucyrus-Erie "Dozershovel" utilizes steel piping and high-pressure flexible hose. Rotary latch-type dumping trip mechanism produces rolling instead of sliding action between members during unlatching. Welded construction is used throughout

Right—Variable speed of Pyrex print cylinder in Bruning printer is effected by V-belt drive of bevel gear-type differential. Input sun gear of differential is driven by adjustable V-pulley, the effective diameter of which is controlled by hand knob through rack and pinion. Final drive from differential output is accomplished through roller chain

Below—Only three gear contacts occur between drive motor and cutter in Kent-Owens vertical miller. Wide face spline-mounted pickoff gears under light aluminum cover on head provide wide range of spindle speeds. Gears run in oil bath and are easily removed for speed changing







Below—A writer and signer operating in tandem constitute Todd's new check-writing machine. Drive from motor to clutch assembly is by belt and bearings utilized are bronze. Shafts are SAE B-1113; type in writer unit is forged brass; signature plates are made of copper

Above — Driven through change gears, fully enclosed lead screw with trapezoidal threads traverses the work table of Societe Genevoise precision thread grinder. Work spindle is case-hardened chrome-nickel steel and runs in a hardened steel and a bronze bearing

Machines Behind the Guns

(For new machine listings, see page 292)

Left—Actuated by velocity of air at constant pressure, the Sheffield size checker employs float in transparent tube for indicating size variations. Air is fed from unit to gaging fixture through flexible rubber hose to enhance utility

Below—Changes in oil viscosity due to temperature variations are prevented from influencing accuracy of hydraulic feed in the Landis plain hydraulic grinder by automatic compensating device. All electric controls are mounted in enclosed compartment at rear of machine bed

Left—Flame adjustments are controlled by needle valves and movement of flame to and from ampule is controlled by bronze cams in Popper & Klein's ampule sealing machine which employs cast iron and machine steel main frame. Gear and belt drives are used

Design Roundup

Lean Alloys Will Stay

EVIDENCE is accumulating that a number of the NE lean alloy steels will continue to be used in large-scale production after the war. Some manufacturers who through necessity have replaced higher alloy compositions formerly considered indispensable likely will find that their products have not suffered from the change and will wish to enjoy the economic advantages resulting from utilization of the leaner compositions. Aside from the price angle, needless waste of domestic or world supply of steelmaking materials cannot continue even during peacetime without serious consequences. Retention of the lean alloy steels probably will be influenced favorably by progress in the comparatively recent efforts to evaluate quantitatively the relative effect of alloying elements on hardenability, as well as in the use of the so-called hardenability intensifiers.

Difficulties in Cooling Electrodes

REFRIGERATION of offset-type resistance welding electrodes, in the welding of special aluminum sections, has been a problem complicated by the difficulty of directing the refrigerant to the tip of the electrodes where it is most essential. This is not so difficult in straight electrodes where the internal "piping" to the tip is a simple matter; in the offset-type electrode the solution has been to arrange the refrigerant lines outside the electrode and braze them to the shank just back of the tip, drilling holes at this point to admit the cooling solution.

Flanged Bearings Aid Production

FLANGE-TYPE ball bearings are being used on control valve shafts of hydraulic torque amplifiers in coastal defense computor mechanisms. In this and many other applications, their use helps materially to facilitate production. When the customary unflanged type of ball bearing is employed, depth of the bearing seats in machine members must be held within close limits. With flange-type bearings, however, since the flange of the bearing itself provides the seat, it is merely necessary to bore or ream a "through" hole in the receiving member. This represents definite savings both in machining time and checking.

Where a number of plates or flat castings are to be machined to receive bearings on identical centers, they can be stacked and the bearing holes bored through in a single setup. Such procedure, ideal for production jobs, is also applicable when plates are to be used in the "breadboard" type of cut-and-try experimental mechanisms familiar to all test and research laboratories.

Although the purchase price of these bearings is necessarily higher than that of the unflanged types, savings realized in machining, checking and assembly often more than offset the original price difference.

Improving Casting Designs

DESIRABILITY of closer working relations between the design engineer and the foundryman is not a new subject but, nevertheless, is one which still stands in need of stressing. Hence, attention to this question at a recent War Production conference in New York proved both timely and helpful.

The advantages of a get-together between design engineer and foundryman before patterns were built and of a greater study of casting practice by the designer, with the help of the foundryman, were laid down as fundamental. A number of suggestions came under the general head of simplification, such as the desirability of having as few subdivisions as possible in castings, and of having no sharp changes in section which invariably result in shrinkage cracks and distortion. Simple cores with provisions for sufficient support were also suggested, with long, slender cores to be avoided. Simple cores, it was pointed out, can often be standardized and machine made, resulting in a cost reduction to the profit of all concerned.

A warning was issued against the use of too large fillets, as they cause shrinkage. Also, a design should always be drawn up with an eye to the possibility that the casting might be placed on a quantity production basis. Permanent information should be specified on drawings, such as relates to soundness (in die castings especially), acid resistance, pressure tightness, points of stress and X-ray examination.

Calculating Hollow Rolls for Minimum Deflection

By R. G. Minarik
Professor of Mechanical Engineering
Syracuse University

DESIGNERS of machines often are confronted with the problem of determining the correct wall thickness of relatively long, uniformly loaded, hollow rolls, *Fig.* 2, to meet an imposed condition of minimum deflection. It generally is recognized that an optimum wall thickness, or ratio of inside diameter to outside diameter, exists which will allow the least deflection of the roll under a given load. However, methods for the determination of this ratio are not found in texts or handbooks and consequently the designer proceeds with his solution by the method of tedious trial calculations.

A direct method for obtaining the inside diameter, the "least maximum" deflection and the maximum induced tensile stress for given values of roll length, outside roll diameter, applied distributed load and physical properties of the roll material is presented in this data sheet. Solution is facilitated by use of the chart, *Fig.* 1, Page 142.

Nomenclature

- L =Length of roll, inches
- D =Outside diameter, inches
- d =Inside diameter, inches
- N $=d/D$, for general equations
- n $=d/D$, for minimum deflection equations
- W =Applied distributed load, pounds per inch
- w =Weight density of roll material, pounds per cubic inch
- E =Elastic modulus of roll material, pounds per square inch
- S =Maximum tensile stress, for general equation, pounds per square inch
- s =Maximum tensile stress, for minimum deflection equation, pounds per square inch
- Y =Maximum deflection, for general equation, inches
- y ="Least maximum" deflection, inches.

Using the notation given in the Nomenclature, the general equation for the maximum deflection of a hollow roll loaded as shown in *Fig.* 2 may be written as:

$$Y=\frac{L^4}{4.8ED^2}\left[\frac{4W}{\pi D^2}\left(\frac{1}{1-N^4}\right)+w\left(\frac{1}{1+N^2}\right)\right] \quad ..(1)$$

and the general equation for the maximum tensile stress induced by this deflection as:

$$S=\frac{L^2}{D}\left[\frac{4W}{\pi D^2}\left(\frac{1}{1-N^4}\right)+w\left(\frac{1}{1+N^2}\right)\right] \quad ....(2)$$

The problem of determining optimum wall thickness—that which will give a least value for the maximum deflection Y of Equation 1—resolves itself into the finding of the optimum value of N. This N will be designated by the symbol n. The corresponding "least maximum" deflection will be denoted by the symbol y.

Differentiating Equation 1 with respect to N, substituting the notation $n=N$ and setting the result equal to zero, yields

$$\frac{W}{wD^2}=\frac{\pi}{8}\left(\frac{1-n^2}{n}\right)^2 \quad ....(3)$$

The value of n, for given values of W, w and D, determined from Equation 3 and substituted into the general equations establishes the optimum inside diameter, the "least maximum" deflection and the maximum induced tensile stress. The relations are given in Equations 4 to 10 on Page 143.

Note that Equations 4 to 10 hold only for values of n consistent with Equation 3. Equations 1 and 2 have general application for any assigned value N between zero and one, but will yield values of "least maximum" deflection only when $N=n$.

A direct evaluation of n from Equation 3 is somewhat awkward. However, the use of the curves on *Fig.* 1 reduces the determination of n, as well as y and s, to the simple routine procedure outlined on Page 143.

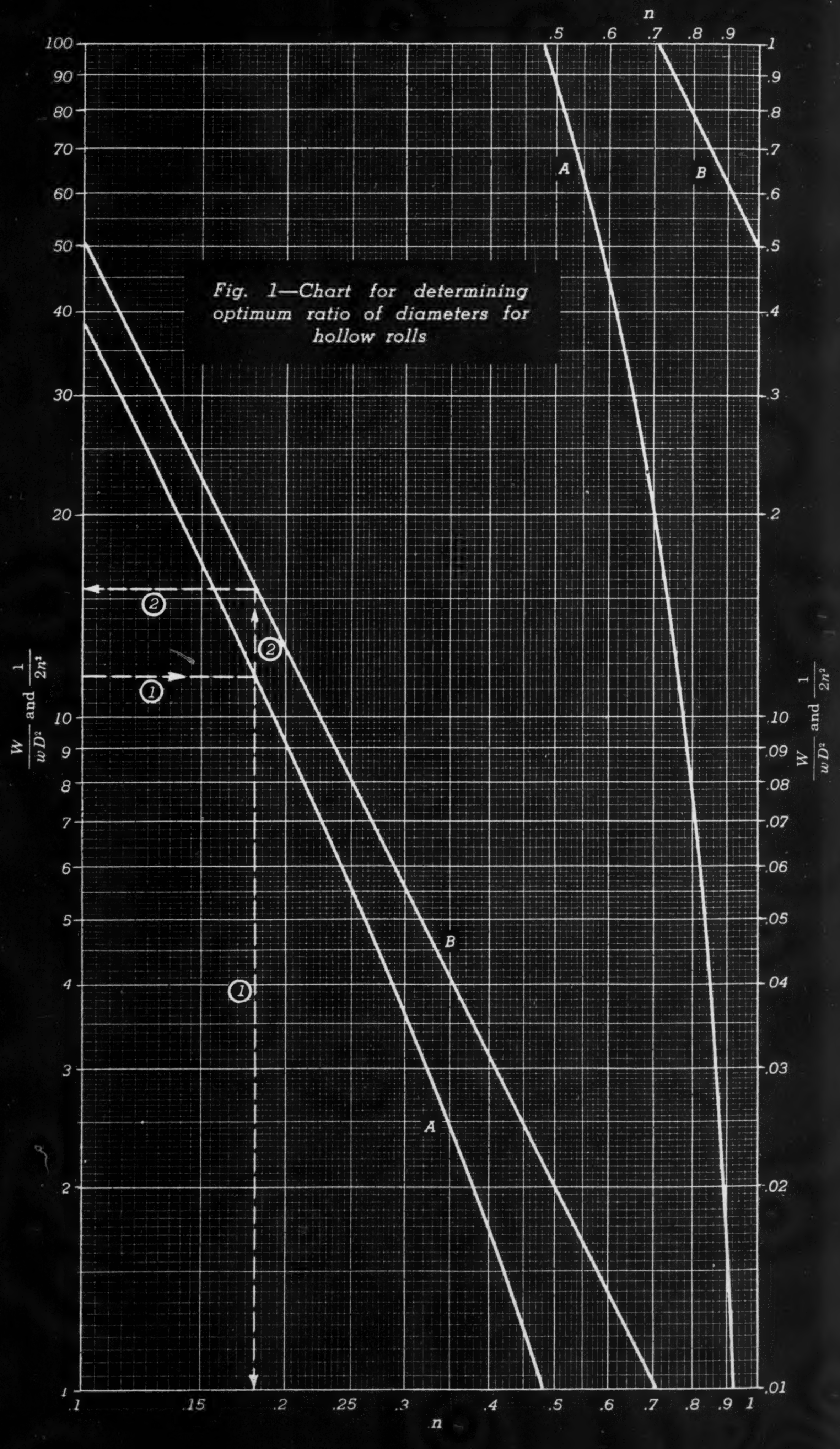


Fig. 1—Chart for determining optimum ratio of diameters for hollow rolls

Procedure

1. Calculate W/wD^2. For common materials, w is given in the accompanying table.

2. Enter chart, *Fig.* 1, with this value of W/wD^2 on the appropriate vertical scale, project horizontally to the nearest A-curve, then vertically to one of the n-scales for the value of n. Example: $W/wD^2 = 11.5, n = .18$.

3. From the point on the A-curve just found, project vertically (either upward or downward) until a B-curve is intersected, then horizontally to the related vertical scale for the value of $1/2n^2$. Example: $1/2n^2 = 15.5$.

4. Use these values to calculate inside diameter d, least maximum deflection y and the maximum induced tensile stress s from Equations 4 to 10 below, selecting the proper formulas according to the conditions of end fixity shown on *Figs.* 2, 3 and 4.

Density and Elastic Modulus

Material	w lb./cu. in.	E psi
Steel	.283	30,000,000
Cast iron	.260	10,000,000
Cast brass	.306	12,500,000
Hard drawn brass	.306	15,000,000
Gunmetal	.318	16,000,000
Mg 10% Al 90%	.072	10,300,000

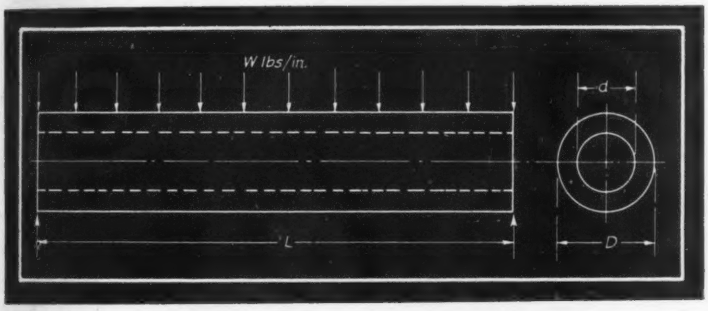


Fig. 2—Uniformly loaded roll, freely supported ends

$$d = nD \qquad (4)$$

$$y = \frac{L^4 w}{4.8ED^2}\left[\frac{1}{2n^2}\right] \qquad (5)$$

$$s = \frac{L^2 w}{D}\left[\frac{1}{2n^2}\right] \text{ (at center)} \qquad (6)$$

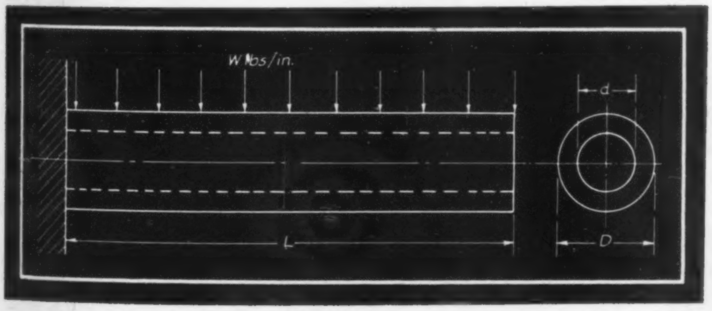


Fig. 3—Uniformly loaded roll, cantilever support

$$d = nD \qquad (4)$$

$$y = \frac{2L^4 w}{ED^2}\left[\frac{1}{2n^2}\right] \qquad (7)$$

$$s = \frac{4L^2 w}{D}\left[\frac{1}{2n^2}\right] \text{ (at support)} \qquad (8)$$

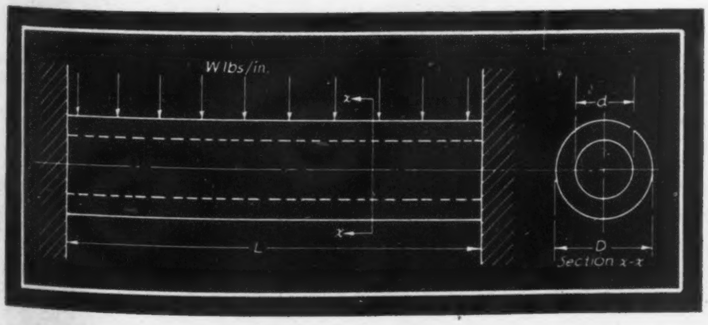


Fig. 4—Uniformly loaded roll, fixed ends

$$d = nD \qquad (4)$$

$$y = \frac{L^4 w}{24ED^2}\left[\frac{1}{2n^2}\right] \qquad (9)$$

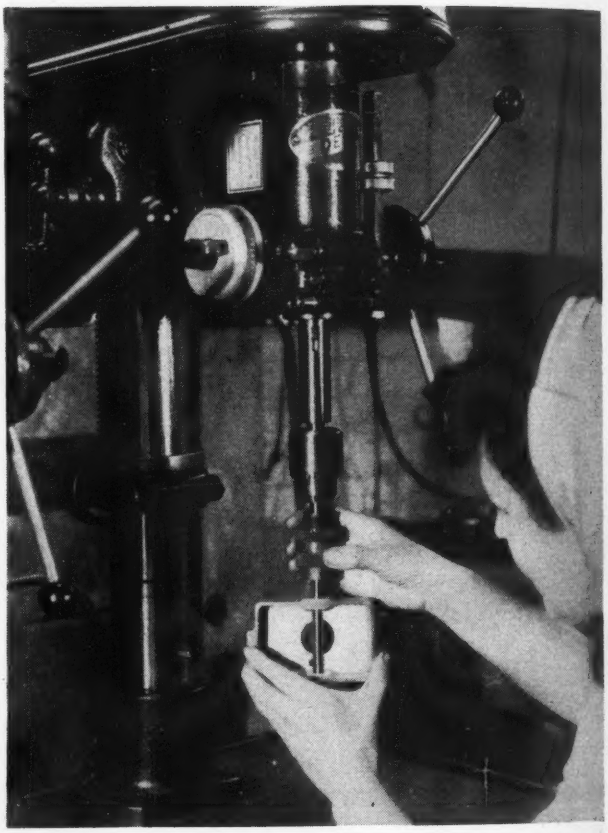
$$s = \frac{2L^2 w}{3D}\left[\frac{1}{2n^2}\right] \text{ (at support)} \qquad (10)$$

Applications

of Engineering Parts, Materials and Processes

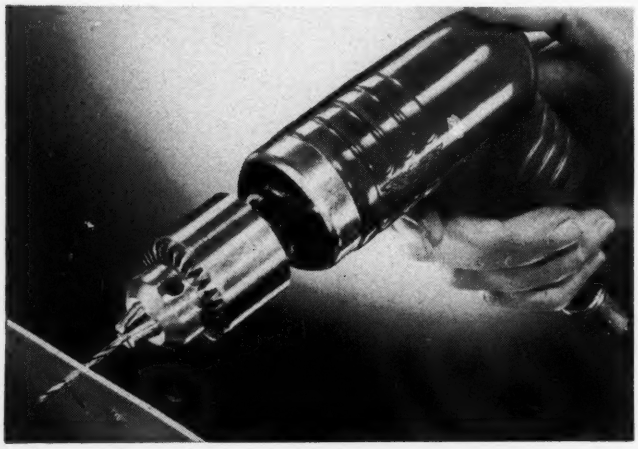
Chrome Plating for Wear Resistance

GREATLY increased wear resistance offered by hard chrome plating has been utilized to prolong the life of pilot shafts on Nobur chamfering tools, right. Used in a drill press, lathe or other machine spindle, the burring tool rotates continuously while parts are fed to it as fast as the operator can handle them. The chrome treatment has been found particularly effective in resisting the abrasive action of hard aircraft materials.



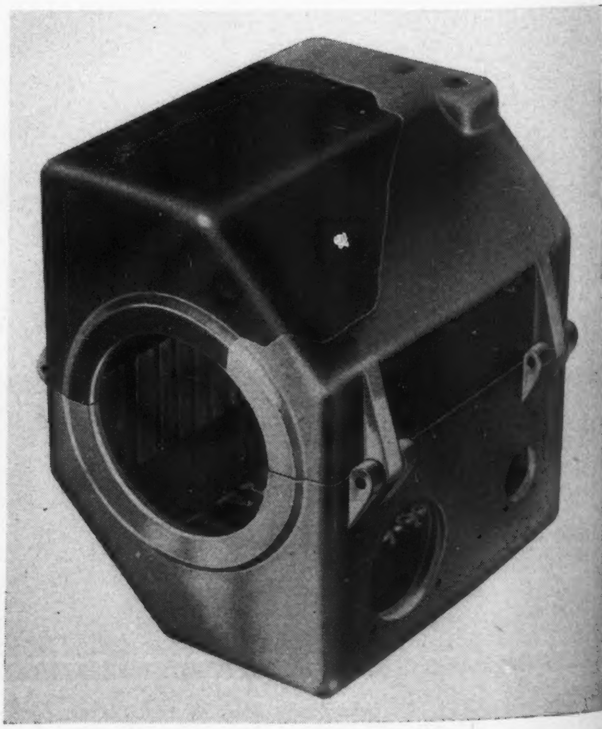
Plastic Drill Housing

LIGHTWEIGHT plastic housings on Aro rotary pneumatic drills, below, enable women employees to handle man-size drilling jobs with more accuracy and less fatigue. A natural-fit grip and smooth contour, as well as decreased weight, make the drill comfortable to handle for continuous working. Made of Tenite, the housings possess high impact strength and resilience and will not chip or dent, while hard blows on the housing are not likely to damage the operating parts of the drill.



Electroplating on Plastics

SURFACES of the plastic swivel joint housing shown at right are electroplated with copper and cadmium for purposes of electrostatic shielding. The metal coating, which is deposited by the Metaplast process, is smooth, nonporous and maintains the detail of the underlying surface. Through use of plastics considerable weight is saved as well as the time required to fabricate intricate metal housings, while the strongly adhesive metal coating furnishes desirable properties such as high conductivity, resistance to vibrational stresses, and dimensional stability.



Materials Work Sheet

Filing Number 14.00

Monel

WROUGHT AND CAST

AVAILABLE IN: Cold-drawn, hot-rolled and forged rods and bars; cold-drawn wire; hot-rolled plates; cold-rolled sheets and strips; cold-drawn seamless tubing; welded tubing; castings.

ANALYSES:

	Ni	Cu	Fe	Mn	Si	C	S
Wrought	67	30	1.4	1	.1	.15	.01
Cast	67	29	1.5	.9	1.25	.3	.015

PROPERTIES

TENSILE STRENGTH

(*psi*)

Cold-drawn rod and bar, annealed	70- 85,000
as drawn	85-125,000
Hot-rolled rod and bar	80- 95,000
Forged rod and bar	80-110,000
Cold-drawn wire, annealed	70- 85,000
No. 1 temper	90-110,000
regular temper	110-140,000
spring temper	140-170,000
Hot-rolled plate, as rolled	80-110,000
annealed	70- 85,000
Standard cold-rolled and annealed sheet and strip	65- 85,000
Special cold-rolled sheet, hard temper	110-120,000
Special cold-rolled strip, full hard temper	100-140,000
No. 35 Finish sheet	78- 85,000
Cold-drawn seamless tubing, annealed	65- 85,000
as drawn	90-125,000
Cast monel	65- 90,000

YIELD STRENGTH

(*.2% offset, psi*)

Cold-drawn rod and bar, annealed	25- 40,000
as drawn	60-120,000
Hot-rolled rod and bar	40- 65,000
Forged rod and bar	40- 85,000
Cold-drawn wire, annealed	25- 40,000
No. 1 temper	50- 85,000
regular temper	85-130,000
spring temper	130-160,000
Hot-rolled plate, as rolled	40- 90,000
annealed	25- 45,000
Standard cold-rolled and annealed sheet and strip	25- 45,000
Special cold-rolled sheet, hard temper	90-110,000
Special cold-rolled strip, full hard temper	90-130,000
Cold-drawn seamless tubing, annealed	25- 35,000
as drawn	60-120,000
Cast monel	33- 40,000

ELONGATION IN 2 INCHES

(*per cent*)

Cold-drawn rod and bar, annealed	50-35
as drawn	35-15
Hot-rolled rod and bar	45-30
Forged rod and bar	40-20
Cold-drawn wire, annealed	50-30

MACHINE DESIGN is pleased to acknowledge the collaboration of The International Nickel Company, Inc. in this presentation.

ELONGATION (*cont'd.*)

Cold-drawn wire, No. 1 temper	20- 5
regular temper	15- 4
spring temper	10- 2
Hot-rolled plate, as rolled	45-20
annealed	50-30
Standard cold-rolled and annealed sheet and strip	50-20
Special cold-rolled sheet, hard temper	15- 2
Special cold-rolled strip, full hard temper	15- 2
Cold-drawn seamless tubing, annealed	50-20
as drawn	20-10
Cast monel	45-25

BRINELL HARDNESS

(*3000 kg*)

Cold-drawn rod and bar, annealed	110-140
as drawn	160-250
Hot-rolled rod and bar	140-185
Forged rod and bar	140-220
Hot-rolled plate, as rolled	140-220
annealed	110-140
Cast monel	125-150

ROCKWELL B

Standard cold-rolled and annealed sheet and strip	61-73
Special cold-rolled sheet, hard temper	94 min.
Special cold-rolled strip, full hard temper	98 min.
No. 35 Finish sheet	74-89

ELEVATED TEMPERATURE PROPERTIES OF HOT ROLLED RODS

(*short-time tests*)

Temperature Deg. F.	Tensile Strength (psi)	Yield Strength (psi)	Elongation in 2 inches (per cent)
70	81,000	32,000	46
200	79,000	29,000	46
400	78,000	26,000	44
600	78,000	28,000	51
800	71,000	29,000	52
1000	51,000	23,000	29
1200	30,000	18,000	34
1400	22,000	13,000	46
1600	13,000	......	54
1800	8,000	......	45

Materials Work Sheet

IZOD IMPACT

(ft-lb)

Cold-drawn rod and bar, annealed	120- 90
as drawn	115- 75
Hot-rolled rod and bar	120-100
Forged rod and bar	115- 75
Cast monel	80- 65

LOW-TEMPERATURE PROPERTIES OF FORGED RODS

Temperature Deg. F.	Yield Strength .2% offset (psi)	Tensile Strength (psi)	Elongation in 2 inches (per cent)	Reduction of Area (per cent)	Izod Impact (ft-lb)
FORGED					
70	67,000	92,000	31.0	72.7	119
—297	91,500	128,250	44.5	71.8	119
—423	96,400	142,000	38.5	61.0	...
ANNEALED					
70	31,300	78,650	51.5	75.0	100
—297	49,500	115,250	49.5	73.9	119

CREEP STRENGTH OF HOT-ROLLED RODS

Temperature Deg. F.	Stress to Produce .1% Elongation in 10,000 hours	Stress to Produce 1% Elongation in 10,000 hours
750	20,000	31,000
800	15,000	23,500
900	.....	13,000

PHYSICAL CONSTANTS

Specific Gravity	8.84
Weight (lb. per cu in)	.319
Melting Point (deg. F.)	2460
Specific Heat (80-750 deg. F.)	.127
Coef. of Thermal Expansion per deg. F. (80-212 deg. F.)	.0000078
Thermal Conductivity (g-cal/sq cm/sec/deg C/cm) 0-100 deg. C.	.062
Electrical Resistivity (ohms per circular mil per ft at 68 deg. F.)	290
Modulus of Elasticity in Tension	26,000,000
Modulus of Elasticity in Torsion	9,500,000
Poisson's Ratio	.32

APPLICATIONS

Monel is particularly suitable for highly stressed parts which require toughness and exceptional resistance to corrosion. It is extensively used for such parts as propeller shafts, pump shafts and rods, valves and impellers, bolts, rivets and other standard fastenings, steam turbine blading, gaskets, steam traps, orifice plates, meter parts, high-pressure feedwater heater tubing, tanks, electrical contact parts, springs, carburetor parts, strainers, screens and filters.

In the chemical process industries, handling such products as chemicals, pharmaceuticals, foodstuffs, beverages, paints and varnishes, soap, dyestuffs, paper, rayon, etc., this metal finds wide application for such parts as autoclaves, stills, mixers, heat exchangers, coolers, condenser coils, evaporators, fittings, conveyors, chutes, hoppers, rolls, doctor blades, spray nozzles, etc.

CHARACTERISTICS

Monel is a "general purpose" rather than a "specialized" alloy and combines high strength, ductility and excellent resistance to corrosion. Its characteristics are retained through manufacturing and fabrication operations and throughout its service life. Under corrosive conditions, its cold-worked, welded and cast forms behave substantially the same. Its strength range is appreciably above that of the common types of brasses, bronzes and nickel-silvers, comparing favorably with that of some alloy steels. While it cannot be hardened or made stronger by heat treatment*, its strength can be increased by mechanical (cold) working. Monel's high ductility is demonstrated by the fact that the spring-temper wire can be bent around its own diameter without fracturing. Commercially the most important of the nickel-copper alloys, monel is unique in that the nickel and copper of which it is composed are refined from the ore without separation. Thus, it is known as a "natural alloy".

*"K" monel, an alloy of substantially the same composition as monel to which has been added 2.75% aluminum, is an age-hardenable alloy that can be hardened by heat treatment.

FABRICATION

MACHINABILITY:

Because monel is exceptionally tough, it requires the use of high-speed or cemented carbide tools ground to sharp cutting edges, ample flow of cutting compound and moderate speeds and feeds. Where parts must withstand corrosive action, a good smooth finish should be specified, as rough, torn surfaces often provide focal points for acceleration of corrosion. Cold-drawn rods and bars in the "as-drawn and normalized" temper will machine best. For automatic screw machine and similar operations, "R" monel, a machining quality rod, should be specified. However, if the rods are to be subjected to cold-forming operations, annealed or No. 1 temper should be specified. When specifying hot-rolled rounds, flats, squares and hexagons for machining, allowance should be made in overall size to insure that the finished size will have properly cleaned surfaces. Recommended allowances are here listed:

Allowances for Finish Machining of Hot-rolled Products

Size (inches)	Allowance for Rounds	Allowance for Flats, Squares and Hexagons
Up to 7/8 incl.	1/8-in. on diam.	1/8-in. on each side
1 to 1 7/8 incl.	1/8-in. on diam.	3/16-in. on each side
2 to 2 7/8 incl.	1/8-in. on diam.	3/16-in. on each side
3 to 4 1/2 incl.	3/16-in. on diam.	3/16-in. on each side

DRILLING:

While standard twist drills as furnished by drill manufacturers are satisfactory, they will function most efficiently with polished flutes. Such drills, made of high-speed steel of the 18-4-1 type of tool steel are used extensively and are reground as soon as signs of dulling appear. Drills are kept feeding into the work, for if they are permitted to dwell without cutting, they polish and harden the metal at the base of the hole, making it difficult to resume cutting as well as dulling the drill. Work is flooded with cutting oil during drilling.

REAMING:

Spiral-fluted, high-speed steel reamers with narrow lands and well polished flutes are generally used and are kept sharp at all times. To obviate the possibility of digging and chattering, ample lubrication with a suitable cutting oil is

employed on all reaming operations. Recommended amounts of material to be removed by reaming in various size holes are listed below:

Amount of Material to be Removed by Reaming

Hole Diam. (inches)	Material to be removed
1/8 to 1/4	.005-in. on diam.
1/4 to 1/2	.008 to .012-in. on diam.
1/2 to 1	.015-in. on diam.
1 to 2	.031-in. on diam.

TAPPING:

Being extremely tough, this material requires a slightly larger tap drill for a given thread than is customarily considered standard. Specify a tap-drill size which will result in 75% depth of thread. Experience shows that such a thread is only 5% weaker than a full-depth thread. In addition, monel, having high ductility, will flow into the roots of the tap threads, producing what for all practical purposes amounts to a full-depth thread. A suitable lubricant is always used in tapping operations since without lubrication monel has a tendency to gall on the tap.

PERFORATING:

Because of the toughness of monel, the minimum hole diameter that can be punched, on an economical production basis, must be larger than the thickness of the sheet up to thicknesses of approximately 5/32-inch. For heavier plate, the minimum diameter of the hole may be equal to the thickness of the sheet. Recommended minimum hole diameters for various sheet thicknesses are listed below and apply to soft and quarter-hard tempers:

Minimum Punched-Hole Diameters

Sheet Thickness (inches)	Min. Hole Diam. (inches)
.018 to .034 incl.	Sheet thickness x 1.5
.037 to .070 incl.	Sheet thickness x 1.3
.078 to .140 incl.	Sheet thickness x 1.2
5/32-inch and heavier	Sheet thickness x 1

The hole diameters listed above are only general recommendations. In particularly favorable setups, it is possible to punch holes smaller than the sizes listed. Die and punch clearances for light-gage sheets are the same as for neat steel punching. For 1/4-inch and heavier plate, less clearance than for steel is often used to produce clean holes, free from burrs. A heavy, sulphur-base oil is used as a lubricant in punching and is thoroughly removed before the punched sheet is annealed or heated for any other purpose.

DEEP DRAWING:

Cold-rolled sheets and strips in soft temper and with fine to medium grain are most suitable for deep drawing. Monel responds well to cold working and can be formed into any shape that can be produced from steel of deep-drawing quality. Very few commercial operations require changes or variations in die equipment designed originally for drawing steel or brass. Draw-ring and punch-nose radii for light-gage cylindrical shells are usually from 5 to 12 times the stock thickness. Corner-edge radius for a rectangular shell should be from 4 to 10 times the stock thickness. Because deep drawing cold works the material, annealing may have to be resorted to between successive draws (see Annealing). Suitable lubricants are used on all deep-drawing operations.

SPINNING:

Monel requires more power in spinning than do softer metals such as copper, brass and aluminum. Also, since it

work hardens more rapidly, it requires more frequent annealing than the copper-base alloys and aluminum. In general, roller-type tools are preferred because they create less friction than do the solid-nose types. Monel is spun at speeds from one-half to three-quarters of those used for the softer metals, and a heavy-bodied lubricant is always used.

WELDING:

Electric arc, oxyacetylene (gas) and resistance welding are the types most predominantly employed on monel. The oxyhydrogen flame is not practicable for the welding of monel because of its low temperature. Hammer welding is not possible because fusion does not occur unless the metal is melted. After welding, no thermal or chemical (passivation) treatments are necessary or recommended to retain or restore corrosion resistance. Since the coefficient of expansion is practically the same as for steel, warping and buckling resulting from a welding operation is essentially the same as for a similar construction of steel. Electric welding, if applicable, will produce less buckling than gas welding.

SILVER BRAZING:

Strength and corrosion resistance of properly designed and executed silver-brazed joints are greater than those of soft-soldered joints. Strength of a properly made silver-brazed joint should be approximately 50,000 pounds per square inch in tension and 30,000 pounds per square inch in shear at room temperature. These values can be used in design calculations with a factor of safety of five. Maximum temperature to which silver-brazed joints can be expected to carry an appreciable load is 600 degrees Fahr. and at that temperature their tensile strength is about 50 per cent of its value at room temperature. Shear (lap) type joints are used more widely than butt joints because of the ease of maintaining better fits and the facility wherewith adequate strength can be obtained merely by extending the lap-shear area.

CORROSION RESISTANCE

Monel is resistant to most acids, alkalies, salts, waters, food products, organic substances and atmospheric conditions, both at normal and elevated temperatures. In sea water, it is particularly useful for vital parts where high velocity effects are encountered. However, where exposure in sea water permits accumulation of marine growths or other deposits, some local attack may occur. Monel in indoor atmospheric exposure either remains bright or undergoes slow fogging. In outdoor urban atmosphere, tarnishing is relatively rapid, developing with time into a green coating. In marine atmospheres a layer of fine corrosion product accumulates. Monel is not useful in highly oxidizing acids such as nitric and nitrous, and has only limited usefulness in sulphurous acid except in cold, dilute solutions.

GALVANIC CORROSION

In neutral salt solutions and waters, monel, nickel, copper and high copper alloys may ordinarily be coupled without seriously accelerated attack. In sea water it is wise to avoid couples of yellow brass with monel where the exposed area

Materials Work Sheet

GALVANIC CORROSION

(Continued)

of monel is many times that of the brass. With approximately equal areas, or where the area of the monel is smaller than the brass, such couples are safe, as for example with brass propellers on monel shafts.

ANNEALING

Lubricants and other foreign matter are always thoroughly removed before either box or open annealing. Box annealing is used for small pressed parts, rivets, wire in coils, etc. Open annealing in oven type or reverberatory furnaces is used for cupped or drawn shapes, spinnings, rods and tubing. In annealing, the time-temperature relationship is of prime importance in reference to grain growth. Box annealing is done most satisfactorily at 1350 to 1450 degrees Fahr. for 2 to 6 hours at temperature, the total time in the furnace depending on the rate of heating. The open-annealing range is 1650 to 1800 degrees Fahr. for 2 to 5 minutes at temperature when mechanical work is to follow. Spinning and other manual operations sometimes require up to seven minutes at temperature.

DATA ON STOCK FORMS

Hot-Rolled Rod and Shapes

STANDARD SIZES AND MILL LIMITS

(inches)

Mill Limits	Standard Sizes
ROUNDS (in straight lengths)	
¼ to 1 in 1/16ths	¼ to 2 in 1/16ths
1⅛ to 4¼ in ⅛ths	Over 2 to 4¼ in ⅛ths
4½ max.	4½ max.
HEXAGONS (in straight lengths)	
⅜ to 1 in 1/16ths	⅜ to 2⅛ in 1/16ths
1⅛ to 2⅛ in ⅛ths	
SQUARES (in straight lengths)	
5/16 to ½ in 1/16ths	5/16 to 2 in 1/16ths
⅝ to 1 in ⅛ths	Over 2 to 2¼ in ⅛ths
1¼ to 2 in ¼s	

FLATS (in straight lengths)

Standard Thickness	Standard Width	Mill Limit Thickness	Mill Limit Width
⅛	½ to 4	⅛	½ to 8
3/16	½ to 4	3/16	½ to 10
¼	½ to 4	¼	½ to 10
5/16	½ to 4	5/16	½ to 10
⅜	½ to 4	⅜	½ to 10
7/16 all special		7/16	½ to 10
½	⅝ to 4	½	9/16 to 10
9/16 all special		9/16	⅝ to 10
⅝	¾ to 4	⅝	11/16 to 10
11/16 all special		11/16	¾ to 10
¾	⅞ to 4	¾ to 1	Up to 10
1 1/16 all special		1 1/16 to 2	Up to 8

FLATS—THICKNESS AND WIDTH INCREMENTS

Standard	Mill Limit
Thicknesses ½ to 1 in ⅛ths	⅛ to 2 in 1/16ths
Widths from 1¼ to 2 in ¼s	½ to 4 in 1/16ths
Width from 2½ to 4 in ½s	Over 4 in ⅛ths

Widths under ⅝-inch are not edged and will not have sharp corners.

Cold-Rolled Strip, All Widths

THICKNESS TOLERANCES

Thickness (inches)	Tolerance (inches) Plus	Minus
Up to .012	.000	.0005
Over .012 to .050	.001	.001
Over .050 to .078	.001	.0015
Over .078 to .093	.0015	.0015
Over .093 to .125	.0025	.0025
Over .125 to .156	.003	.003
Over .156 to .250	.004	.004

Cold-Rolled Strip, Slit-Edged

AVAILABLE SIZES

Thicknesses (inches)	Width Range (inches)	Maximum Cut Length
COILED		
.010	1 to 8	12 ft.
.012	1 to 9	12 ft.
.015	1 to 12	12 ft.
.018, .021, .025, .028	1 to 16	12 ft.
.031, .034	1 to 18	12 ft.
.037, .043, .050, .056, .062, .070, .078 .093, .109, .125	1 to 18	18 ft.
STRAIGHT LENGTHS		
.130, .140, .156, .171, .187, .203, .218, .234, .250	1 to 18	18 ft.

Cold-Drawn Rod and Shapes

STANDARD SIZES AND MILL LIMITS

(inches)

Standard Sizes	Mill Limits
ROUNDS (in straight lengths)	
½ to 3 in 1/16ths	½ to 1½ in 1/32nds Over 1½ to 3 in 1/16ths
HEXAGONS (in straight lengths)	
½ to 1¼ in 1/16ths	½ to 2 in 1/16ths
1⅜ to 2 in ⅛ths	
SQUARES (in straight lengths)	
½ to ⅞ in 1/16ths	½ to 2⅛ in 1/16ths
1 to 2 in ⅛ths	

FLATS (in straight lengths)

Thickness	Width
3/16 to ¾ in 1/16ths	½ to ¾ in 1/16ths
¼ to 1 in ⅛ths	⅞ to 2 in ⅛ths

Cold-Rolled Strip, Slit-Edged, All Widths

WIDTH TOLERANCES

Thickness (inches)	Tolerance (inches) Plus	Minus
Up to .075	.007	.007
Over .075 to .100	.009	.009
Over .100 to .125	.012	.012
Over .125 to .250	.015	.015

Cold-Drawn Seamless Tubing

STANDARD SIZES

Wall Thickness (inches)	Outside Diam. (inches)
.025, .028, .032	¼ to 1½ in 1/16ths 1½ to 2 in ⅛ths
.035, .042, .049, .058, .065, .072, .083	¼ to 1½ in 1/16ths 1½ to 5 in ⅛ths
.095, .109, .120, .125, .134	⅜ to 1½ in 1/16ths 1½ to 5 in ⅛ths
.148, .165	7/16 to 1½ in 1/16ths 1½ to 5 in ⅛ths
.180 to .187	9/16 to 1½ in 1/16ths 1½ to 5 in ⅛ths
.203	⅝ to 1½ in 1/16ths 1½ to 5 in ⅛ths
.220	11/16 to 1½ in 1/16ths 1½ to 5 in ⅛ths
.238, .250, .259	⅞ to 1½ in 1/16ths 1½ to 5 in ⅛ths
.284, .300, .312	1 1/16 to 1½ in 1/16ths 1½ to 5 in ⅛ths

MATERIAL DESIGNATIONS

ASTM	ASME	Federal	Navy
HOT-ROLLED RODS B164-41T	....	QQ-N-281-1	46M7(INT)
COLD-DRAWN RODS B164-41T	....	QQ-N-281-1	46M7(INT)
FORGINGS B164-41T	S-54	QQ-N-281-1	46M7(INT)
COLD-ROLLED SHEET B127-41T	S-54	QQ-N-281-1	46M7(INT)
No. 35 SHEET (satin finish) B127-41T	....	QQ-N-281-1	46M7(INT)
COLD-ROLLED STRIP B127-41T	....	QQ-N-281-1	46M7(INT)
HOT-ROLLED PLATE B127-41T	S-54	QQ-N-281-1	46M7(INT)
SEAMLESS TUBING B165-41T	S-54	....	44T38
WELDED TUBING	S-54	....	44T38
COLD-DRAWN WIRE	....	QQ-N-281-1	46M7(INT)
CONDENSER TUBES B163-41T	....	....	....
CAST MONEL	....	QQ-C-551	46M1(INT)

PROFESSIONAL VIEWPOINTS

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

"... opens amazing possibilities"

To the Editor:

I was interested in the series "The Shape of Things To Come", by R. S. Elberty, with particular reference to Part III devoted to complex rectangles. His treatment

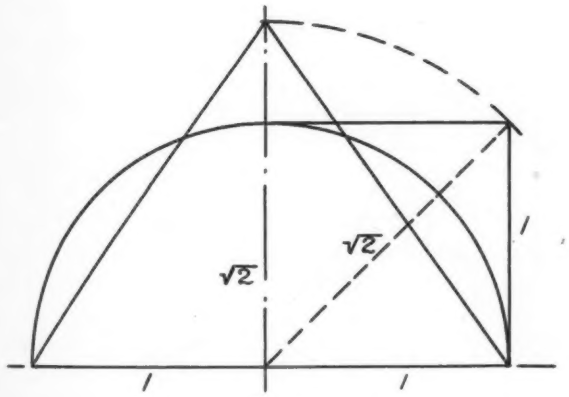


Fig. 1

of the golden rectangle seems to be very adequate. Old artists preferred the golden rectangle as given by the length of members of the human body, whereas nature follows the Lame series in the number of rows of seeds on sunflowers and pine cones, as correctly mentioned by Mr. Elberty. He also mentions the $\sqrt{2}$ but it seems to me that he does not give full justice to this ratio which

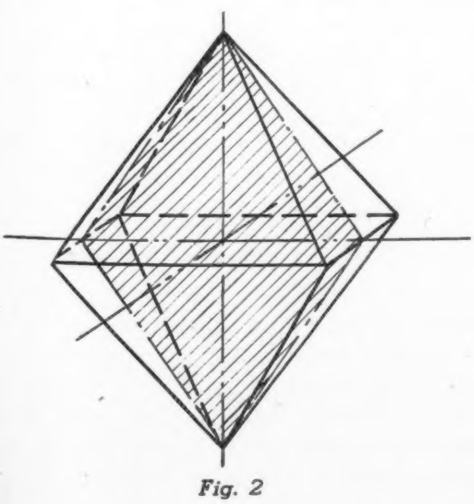


Fig. 2

has amazing possibilities in design.

As it is known, the $\sqrt{2}$ is the diagonal of the square, and is easily found. In architecture the diagonal triangle is in use, *Fig.* 1, in which the base is equal to double length of the square, whereas the height is equal to the diagonal. This triangle when doubled as a rhombus is evidently the section of an octahedron, one of the main regular bodies, *Fig.* 2. This full figure can be inscribed in a rectangle which also in other arrangements (*Fig.* 3) con-

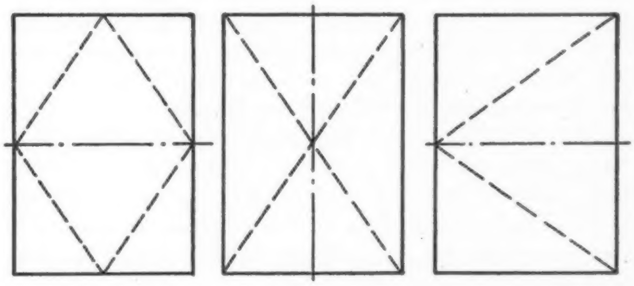


Fig. 3

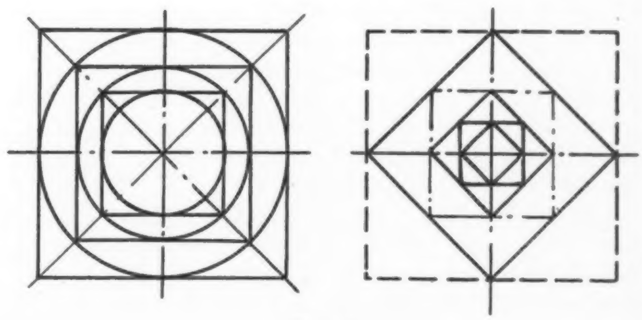


Fig. 4

tains this diagonal triangle. Continual geometric series using the $\sqrt{2}$ are easily obtained as shown in *Fig.* 4 using only squares or squares and circles. Many such figures are found in the drawings of Leonardo da Vinci and other architects.

—PAUL GRODZINSKI
London, England

"... proper selection of materials"

To the Editor:

The new series which commenced in the December issue, "How To Choose the Right Material", is a good supplement to the long series completed in August, "War-

time Metallurgy Conserves Strategic Materials", by R. E. Orton and W. F. Carter. The limitations and possibilities of substitutions have not been given the proper consideration in the past. This is evidenced by the number of permanent substitutions which have been forced upon us by the present emergency, and in many cases the second and third substitutions have proved superior to the primary design. Articles such as those mentioned will contribute much to proper selection of materials.

—W. J. POTTHOFF
Manufacturing Liaison Engineer
The Emerson Electric Co.

" . . . 'hardness' of powder metal parts"

To the Editor:

The article in the November issue on "Designing Powder Metal Machine Parts", by Colin Carmichael, is as good a presentation of the molding phase of powder metallurgy as we have seen to date. However, as with most articles on powder metallurgy, a cardinal sin has again been committed. The statement, "The sintering stage transforms the mass of powder into a hard metal comparable in strength to a casting", is an exaggeration. Powder metallurgists that do the doing, as distinguished from those that do the selling, will agree that the powdered iron part comes out of the furnace comparable in strength to a low-grade cast iron or a white metal casting. But the parts do not come out "hard" if the furnace has done nothing more than sinter the piece. SAE 1020 analysis powder, after it is molded and passed through a commercial sintering furnace having an atmosphere other than bone-dry hydrogen, comes out as "hard" as annealed brass. Furthermore, if the molding has any thin sections, the piece must not be dropped because those thin sections snap off quite easily.

We will grant that it is possible to make iron or bronze structures comparable in strength to a casting of the same analysis by applying one or more metallurgical operations during and after sintering or even prior to heat treatment. These operations are not objectionable from a technical standpoint, but they add costs that affect the selling price. Current literature, however, is prone to describe the wonders of powder metallurgy rather than to discuss the drastic methods necessary to process the parts so that they can compete with fused metal. Possibly the reason why little mention is made of these difficulties is that salesmen do not object to bone-dry hydrogen atmospheres, molding pressures of 100 tons per square inch or more, molding at high temperatures, hot forging, high-pressure coining—not for the purpose of sizing but for the purpose of cold working. The process, as far as some writers are concerned, is always molding and sintering, the implication being that the next step is shipping.

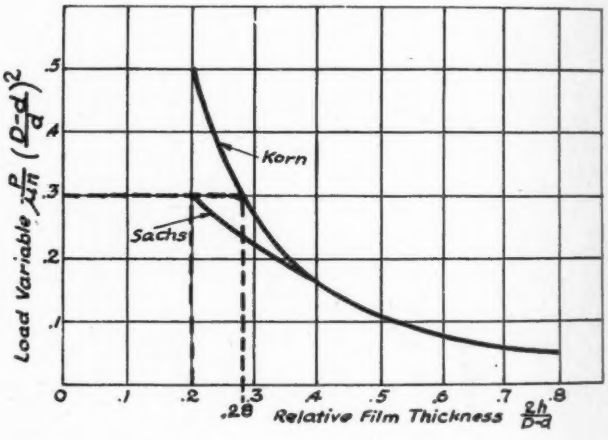
—R. E. STECK, *Manager*
Molded Metals Dept.
Raybestos Div. of Raybestos-Manhattan Inc.

" . . . performance of bearings"

To the Editor:

In the article "Designing Bearings for Fluid-Film Lubrication", by Arthur H. Korn (MACHINE DESIGN, January), *Fig.* 3 shows load variables for sleeve bearings as reported by me, with the comment that it is difficult to say whether they approach actual conditions more closely than do the Kingsbury factors. In *Fig.* 4 the author of the article presents test results for the bearing ratio $B/D=15/11$, which he evidently considers reliable. Comparison of this test curve with calculations I have made for a bearing of these proportions and based on my own findings appears in the accompanying figure. It can be seen that there is full agreement for relative film thicknesses from .8 to .4 and that the disagreement is 15 per cent and 40 per cent respectively at relative film thicknesses of .3 and .2.

What does a discrepancy of this order mean to the design engineer and to the performance of the bearing? Referring again to the accompanying figure, for a load variable equal to .3 the relative oil-film thicknesses are .28 and .20 from the two curves. To the designer who is interested in oil-film thicknesses it is of no importance whether, for instance, the oil-film thickness actually is 280 or 200 micro-inches as long as the bearing runs in the fluid-friction state. Variations of the clearance as a consequence of manufacturing tolerances will cause wider



fluctuations of the theoretical load-carrying capacity.

Of much greater importance than the above-mentioned deviations and discrepancies is the coefficient of friction, for which no data are given in the article. The coefficient of friction determines the power loss in which, as a rule, the designer is interested. Furthermore it is an important factor influencing the bearing temperature which, in turn, determines oil viscosity. In the articles prepared by me (listed under "References" at the end of Mr. Korn's article), correlation factors reconcile to a sufficiently accurate degree experimental determinations of oil-film thickness, load-carrying capacity *and* coefficient of friction.

—BRUNO SACHS
South Orange, N. J.

"With our biggest battles coming up, this emphatically is not the time to divert any substantial quantities of materials, labor or facilities to less essential civilian production. There certainly cannot be any return to volume production of less essential goods until the war picture is a great deal clearer."—*Donald M. Nelson*

ASSETS to a BOOKCASE

Aircraft Production Illustration

By George Tharratt, chief engineer, Adel Precision Products Corp.; published by McGraw-Hill Book Co. Inc., New York; 201 pages, 8½ by 11 inches; clothbound; available through MACHINE DESIGN, $3.50 *postpaid.*

It would seem that the artist has finally found his niche in engineering. Today we find him firmly ensconced in the drafting room with his three-dimensional illustrations proving a valuable supplement to the traditional orthographic-projection type drawings.

This three-dimensional type of production illustration has so competently provided the answer to quick training of inexperienced production personnel during the war, that its value can hardly fail to be recognized in postwar civilian production. Even the experienced draftsman finds intricate orthographic drawings more easily comprehensible when they are augmented by three-dimensional pictures.

There is no question but that new uses for "picture drawings" will be found, both in the engineering department and in the shop. Thus, the wise draftsman will do well to school himself in this new drawing room technique.

Suffice it to say that this book tells about three-dimensional drawing as it is currently used and being developed to enhance the efficiency of personnel from the shop superintendent right down to the workman on the last menial task on the assembly line. Its illustrations—almost myriad in number—are executed in a competent manner and serve to make this fine volume truly attractive.

□ □ □

Materials Testing and Heat Treatment

By William A. Clark and Brainerd Plehn, Rochester Athenaeum and Mechanics Institute; published by Harper & Bros., New York; 132 pages, 6 by 9 inches; clothbound; available through MACHINE DESIGN, $1.75 *postpaid.*

There is much current discussion concerning the actual value of standard materials tests insofar as their results can be interpreted and put to practical use by the designer of mechanical devices. Instances have been cited, for example, regarding the misleading character of impact test conclusions. Again, the relative worth of ultimate as compared to yield tensile values has been debated pro and con for some time.

This book makes no attempt to give the answers to these problems. It does, however, explain the various test procedures and covers such tests as tensile, shear, torsion, hardness, fatigue, ductility, impact, etc. Materials other than metals are dealt with and include timber, rubber, concrete, cement, lubricants, etc. Final chapters describe heat treatment procedures.

While this book is not recommended as "must" reading for the machine designer, it does have its place in that intangible though often valuable sphere which might be called "background knowledge".

□ □ □

Kinematics and Machine Design

By Louis J. Bradford, professor of machine design and George L. Guillet, associate professor of mechanical engineering, Pennsylvania State College; published by John Wiley & Sons, Inc., New York; 355 pages, 5⅛ by 7¾ inches; clothbound; available through MACHINE DESIGN, $3.00 *postpaid.*

An abridgment of two previous texts: "Kinematics of Machines" and "Machine Design", this book succinctly presents practical information on a variety of mechanical motions and discussions dealing with the design of machine members. It should serve well either as a text for abbreviated wartime courses in machine design or as a handy refresher or reference for the engineer.

After classifying and expounding the various motions, types of acceleration and velocity, it deals with linkages, cams and their followers, elliptical trammels, the pantograph and the geneva. Subsequent chapters discuss stress and strain, properties of materials and factors influencing the design of such machine members as bearings, clutches, brakes, couplings, gears, springs, etc.

□ □ □

Plastics Catalog

Published by Plastics Catalog Corp., New York; 989 pages, 8 by 11¼ inches, clothbound. Available through MACHINE DESIGN, $6.00 *postpaid.*

A correlation of significant articles on plastic materials and processes, together with colorfully illustrated catalog data of leading manufacturers in the plastics industry Its section devoted to "plastics in war" is particularly interesting, showing the utilization of plastics for such items as helmet liners, M52 trench mortar fuzes, periscope lens holders, raincoats, electric wire insulation, aircraft nose sections, gun turrets, etc.

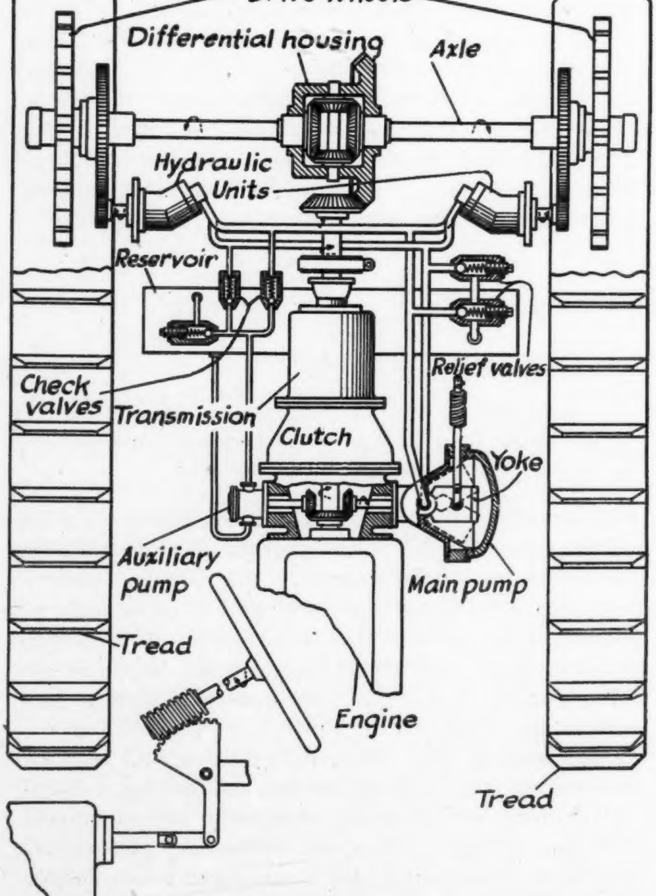
Other sections offer information dealing with tests and specifications, materials, engineering and molding, fabricating, finishing and assembly, machinery and equipment, coatings, synthetic rubber and rubber-like plastics, etc. In addition, numerous folded charts present data on plastics identification, properties, chemical formulas, solvents, plasticizers, etc. Its directory lists educational institutions, producers of materials and chemicals, manufacturers of plastics, molders, tradenames, etc.

Noteworthy Patents

Hydraulics Steers Tractor

AN IMPROVED power transmission system for tractor tread vehicles, in which friction clutches and brakes are eliminated from the steering mechanism and in which positive and precisely graduated control of the relative tread velocity may be maintained, is covered by patent 2,336,911 recently assigned to Vickers Inc. The drive mechanism incorporates hydraulic power transmissions arranged as a supplementary control over a conventional mechanical geared drive.

Principal features of the mechanism are shown in the accompanying illustration. Through a clutch and transmission the engine drives a bevel pinion engaging a ring gear which is part of the differential housing. Drive wheels attached to the two axles may therefore rotate with different velocities when necessary. Geared to each axle are two identical fixed displacement hydraulic units



Hydraulic units geared to each axle function either as pumps or motors. Variable delivery main pump, furnishing surplus fluid to one unit, is used for steering control

which may function either as pumps or as motors. The two ports of the left-hand unit are connected to the two ports of the right-hand unit by a pair of conduits.

In normal operation the engine drives the axles and drive wheels through the differential while the two hydraulic units are driven as pumps through gearing from their respective axles. The left-hand unit discharges oil through one of the conduits to the suction port of the other which, in turn, discharges the oil through the other conduit to the suction port of the left-hand unit. As both units are of the same displacement and are being driven as pumps at the same speed, each fills the capacity requirements of the other. Thus it will be seen that oil will freely circulate between the two units with no pressure drop across either. However, if the left-hand tread, for example, should meet greater resistance than the right-hand tread, the left-hand unit would be unable to pass the delivery of the other unit freely. Under these conditions pressure in one of the conduits will rise and the right-hand unit, acting as a pump, will drive the left-hand unit as a motor, supplying the added torque required for the left-hand tread.

Pump Regulation Controls Steering

Also driven by the engine is a main pump of the variable displacement reversible delivery type having a swinging yoke which is shown in its neutral or zero displacement position. Steering of the tractor is effected through the two hydraulic units, controlled by the stroke regulation of the main pump. Thus when the yoke is moved toward its lowermost position, as shown by dotted lines, the main pump will withdraw oil from one conduit and discharge it into the other. Delivery of the pump plus that of the left-hand unit will exceed the intake capacity of the right-hand unit if the speed of the two units remains the same. Accordingly, pressure in this conduit will rise considerably while the pressure in the other will drop. This pressure drop across the right-hand unit will tend to drive it as a motor in the same direction as it is being driven from the axle but at a faster speed. The left-hand unit, however, will tend to slow down because of the high back pressure in its discharge conduit. Therefore one unit tends to drive its tread in the same direction as it is being driven by the engine, but at a faster speed, while the other unit tends to brake its tread against the drive of the engine.

An auxiliary pump, also driven by the engine, draws oil from the reservoir and delivers it to the low-pressure side of the main hydraulic circuit to make up for losses due to leakage and to the possible opening of the relief valves.

During straightforward travel of the vehicle the driving torque is almost entirely transmitted through the mechanical gearing. Under these circumstances the hydraulic

units act merely as torque balancers and transmit only the difference between the torque required to drive one tread and that required to drive the other. In so doing they act as a positive synchronizing means for the differential axle and insure that the speeds of the two treads are maintained equal.

When the variable pump is shifted out of neutral position for steering purposes, the units are still not required to transmit the full driving torque of the engine but only the relative difference in torque requirement between the two sides of the machine. Thus the hydraulic units required are not as large nor as expensive as if the full driving torque had to be transmitted through them.

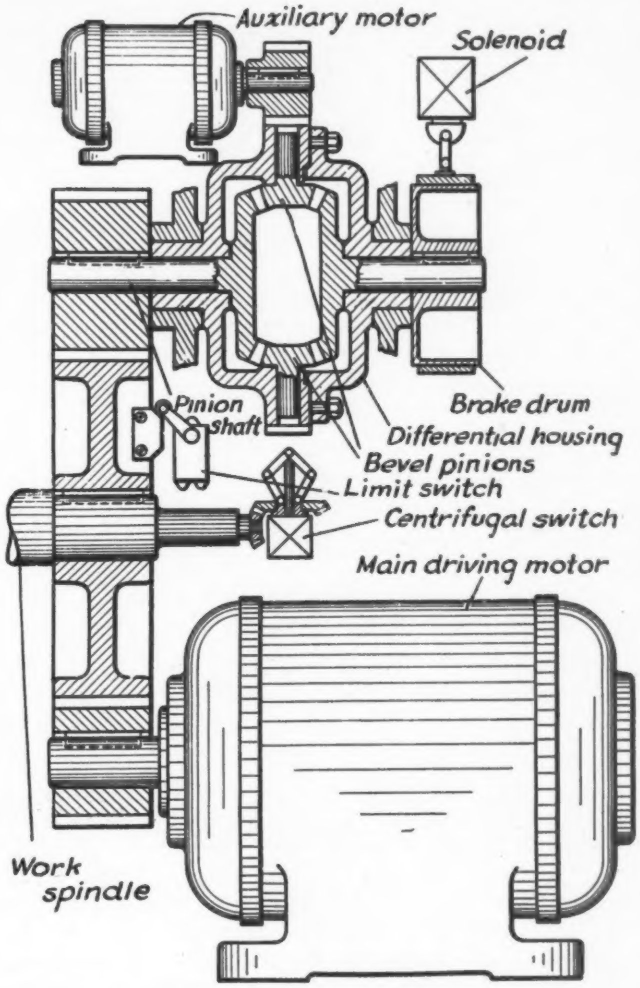
Accurately Stopping a Spindle

SOMETIMES it is desirable to stop the work spindle of a machine tool, at the completion of a cutting cycle, in a predetermined position. Means for accomplishing this rapidly and accurately are covered by patent 2,330,-985 recently assigned to The R. K. Le Blond Machine Tool Co. The device includes an auxiliary slow-speed drive which comes into operation at the completion of the cutting cycle. Control switches for effecting complete stoppage come into action at low speeds, facilitating accurate stopping.

As shown in the accompanying illustration, the work spindle is driven by the main motor through a pinion and gear. Meshing with the gear is a second pinion connected to a differential unit which, in turn, is associated with an auxiliary motor and a friction brake. During normal operation the brake is released and the auxiliary motor deenergized, the differential housing remaining at rest while the brake drum rotates idly in the opposite direction to that of the pinion shaft. As soon as the cutting operation has been completed and the tools are to be retracted away from the work, power is cut off from the main driving motor and the solenoid energized so as to apply the brake. With the brake drum at rest the momentum in the main drive motor and transmission mechanism from the work spindle is transmitted to the differential housing and to the auxiliary motor. Because of the high speed-up ratio between the work spindle and the auxiliary motor, considerable energy is absorbed in accelerating the differential housing and motor armature, thus decelerating the work spindle.

Auxiliary Motor for Slow Speed

When the work spindle speed has dropped to a predetermined value, a centrifugal switch connected to the work spindle energizes the auxiliary motor. The work spindle now continues to rotate at slow speed through the driving action of the auxiliary motor, until the limit switch is tripped by a dog on the work spindle gear, deenergizing the solenoid and releasing the brake. As a result of the instantaneous release of the brake, the pinion shaft stops without jar or coasting action even though the auxiliary motor continues to operate. Under these conditions the driving power from the auxiliary motor expends itself in free rotation of the brake drum and will not tend to rotate the "harder" drive of the main motor and work



Auxiliary motor, brake and differential transmission are used to stop work spindle at end of cutting cycle

spindle. As a result of having the release of the brake effect the accurate stopping, no uncertain frictional problem is involved and coasting is obviated since the spindle driving mechanism is rotating at such a slow speed.

What Motor Truck Operators Want

REPORT of a survey of the opinions of motor truck operators revealed to the Society of Automotive Engineers the following desires: Cooling systems which really cool, maintain their efficiency and are readily serviced; accessibility of critical parts to facilitate and encourage maintenance; electrical systems of high quality; easily-removed unit power plants, especially in large vehicles; provision for quick-starting and rapid warming-up in cold weather; truck cabs which assure comfort and convenience for full-grown drivers and which contribute to safety by eliminating blind spots, cramped driving positions and untimely fatigue; bumpers at standard height so that bumpers alone bump; design improvements to prevent tractor-semitrailer combinations from upsetting in accidents; fewer decorations, less brightwork; and muffling of operating noises to improve motor truck public relations.

Men of Machines

NOMINATION OF Ralph J. Teetor as president of the American Foundrymen's association was recently announced at the meeting of the nominating committee. Mr. Teetor, who is president of Cadillac Malleable Iron Co., is a mechanical engineering graduate of Purdue university. After graduation he became associated with Link-Belt Co., and in 1915 resigned as chief engineer to become secretary of the Standard Malleable Iron Co. During the last war he served as major until December, 1918, and as division inspector until 1919 when he was discharged. He then joined the Howe Chain Co. as vice president and general manager, remaining there until 1921 when he became interested in the iron and lumber business in Cadillac, Mich. His activities in that locality led him to establish the Cadillac Malleable Iron Co., of which he became secretary and general manager. Later he was elected president. Mr. Teetor has been a member of the advisory group and other committees of the foundrymen's association, and has presented papers before annual meetings. He also served as director.

WITH A BACKGROUND in the design and development of clocks and timing systems of various types, Donald E. Anderson is well fitted for his new position as chief engineer of J-B-T Instruments Inc. Prior to his appointment Mr. Anderson spent a major part of his time in the development of vibrating reed frequency meters, electrical testers, etc. While in this position he developed an electron tuning fork frequency standard with associated equipment, for use in calibrating frequency meters. Several years before becoming connected with the J-B-T organization he spent a year at Thomas T. Gibbs Co. working in the development laboratory on timing systems, simplifying production of clocks and watches. He later was in charge of the Guilford Time Laboratory, a branch of New Haven Clock Co. While here he was engaged in further studying systems and improved production of balance wheels. He resigned from New Haven Clock Co. to become development engineer at J-B-T Instruments.

IN HIS NEW CAPACITY with the Kaydon Engineering Corp., B. M. Staley will be able to utilize his wide experience in the machinery and parts fields. For twelve years previous to his appointment Mr. Staley had been with the rotary pump division of National Transit Pump & Machinery Co., serving successively as chief engineer, superintendent and more recently as general manager. While with this company he developed a number of new products besides exercising general supervision over development, engineering, design, production and administration. Mr. Staley is a graduate of the Virginia Military institute, the Carnegie Institute of Technology with a mechanical engineering degree, and Westinghouse Technical school. Early in his career he obtained knowledge in several fields including automotive and stationary gasoline engines, shearing, punching, straightening, rolling and molding machines, etc. In these capacities he also was engaged in reorganization of engineering and management activities. In 1925 Mr. Staley was made manager of the Pittsburgh Machine Tool Co., as well as

vice president of its subsidiary Curtis Pump Co. He was in charge of all operations including engineering. From there, in 1931, he went to National Transit Pump & Machinery Corp., and then on to his present position as factory manager of Kaydon Engineering Corp.

♦

L. S. SHELDRICK has resigned as chief engineer of Ford Motor Co., and is succeeded by D. ROEDER, his former assistant.

♦

ARTHUR W. WEEKS has joined the staff of Stevens-Arnold Co., South Boston, Mass., as special instrument development engineer. Prior to this appointment Mr. Weeks was connected with Mico Instrument Co., Cambridge, Mass.

♦

CLIFFORD A. HARVEY, previously vice president in charge of research and development at Harvey-Wells Communications Inc., Southbridge, Mass., has been made vice president and chief engineer.

♦

A. W. KATH has been promoted from development engineer to chief engineer of Arthur Colton Co.

♦

GILL ROBB WILSON has been elected to a fourth term as president of the National Aeronautical association.

♦

STANLEY PAUL CLURMAN, experimental test engineer, has been made experimental design engineer of the Curtiss Propeller division, Curtiss-Wright Corp., Caldwell, N. J.

♦

E. J. HORTON, formerly chief engineer of Ruud Mfg. Co., has been appointed assistant to the president of Robertshaw Thermostat Co., Youngwood, Pa.

♦

ALBERT R. CROCKER, ex-development engineer at Engineering & Research Corp., Riverdale, Md., has been named assistant chief engineer in the company's propeller division.

♦

JOHN R. BOND has left Studebaker Corp., South Bend, Ind., to join Aerojet Engineering Corp., Pasadena, Calif., as design engineer.

♦

L. W. GUIN, formerly chief engineer, has been made general manager of South Bend Current Controller Co., South Bend, Ind.

♦

VANNEVAR BUSH has been awarded the 1943 Edison medal of the American Institute of Electrical Engineers "for his contribution to the advancement of electrical engineering, particularly through the development of new applications of mathematics to engineering problems, and for his eminent service to the nation in guiding the research program." Dr. Bush is president of the Carnegie Institution of Washington and director of the Office of Scientific Research and Development of the Office of Emergency Management, Washington.

♦

PETER H. VOORLAS, former electrical engineer at Mall Tool Co., is now chief engineer of Electric Motor Corp., Racine, Wis.

♦

LINN L. COLLINS is the new chief engineer at Allied Aviation Corp., Dundalk, Md.

♦

T. C. LEAKE, formerly of the engineering department, Eclipse Aviation division of Bendix Aviation Corp., Bendix, N. J., has joined the engineering department of Graham-Paige Motors Inc., Detroit.

♦

COMMANDER J. G. GOODENOUGH, internal combustion engineer, will head the newly created diesel and engine equipment division of Adel Precision Products Corp., Burbank, Calif.

♦

C. W. VAN RANST has joined the F. L. Jacobs Co., Detroit, as development engineer. Mr. Van Ranst had formerly been chief aeronautical engineer for Ford Motor Co.

♦

HERBERT L. RAWLINS has been made manager of the protective devices engineering department of Westinghouse Electric & Mfg. Co. Mr. Rawlins previously was assistant manager of circuit breaker and protective devices engineering.

♦

TED A. WELLS, in addition to his previous position as chief engineer of Beech Aircraft Corp., Wichita, Kans., becomes vice president of the company.

♦

T. R. RIDEOUT has joined Watson-Flagg Machine Co., Paterson, N. J., as project engineer on worm gear speed reducers and gearmotors. Formerly he had been associated with the Nuttall Works of Westinghouse Electric & Mfg. Co.

♦

A. RICHARD LALLONE has been appointed assistant manager and chief engineer of the aerial navigation products division of Federal Telephone & Radio Corp., Newark, N. J. He formerly had been chief design engineer.

♦

JOHN A. FLANZER now holds the title of general manager in addition to that of chief engineer at Electro Motive Co., Willimantic, Conn.

wherever a tube is used...

For example—

Resistance Welding

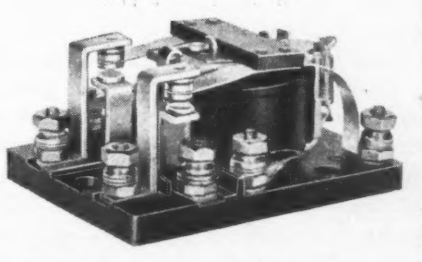
Thyratron tubes, working with other thyratron or ignitron tubes and usually a relay, control the current for spot, projection, seam and other types of resistance welding for lower maintenance and better welds.

THERE'S A JOB FOR

Relays BY GUARDIAN

Your post-war product must stand the competition of *price* as well as quality. And manufacturers who use electron tubes to boost production, cut material costs, and increase product performance, have the edge on competitors. Electronic control of resistance welding is one cost-saver to consider.

In this, as in most other tube applications, the use of a relay increases efficiency. The Series 175 DC and Series 170 AC Relays by Guardian, when used in the output of the tube circuit, control external loads in accordance with the tube operating cycle. These relays have binding post terminals in place of solder lugs. Bakelite bases, molded to reduce surface leakage, give a higher breakdown factor. Contact capacity: 12½ amps., at 110 volts, 60 cycles, non-inductive. Information on contact combinations, coil voltages, and further data is yours for the asking.



Consult Guardian wherever a tube is used. *However,* Relays by Guardian are *NOT* limited to tube applications but may be used wherever automatic control is desired for making, breaking, or changing the characteristics of electrical circuits.

GUARDIAN ELECTRIC

1601-C W. WALNUT STREET CHICAGO 12, ILLINOIS

A COMPLETE LINE OF RELAYS SERVING AMERICAN WAR INDUSTRY

Design Abstracts

Improving Antiknock Fuels

A FACTOR which should be given careful attention when discussing octane number improvements is the well known characteristic of the octane number scale in which, as the numbers become larger, each unit becomes more and more significant. For example, for the interval 69 to 70 octane number one unit improvement may be considered as permitting an increase of .5 per cent in thermal efficiency. In the range 79 to 80 octane number the value of one unit becomes 1.25 per cent in thermal efficiency, while the change from 99 to 100 is accompanied by a potential increase of 3 per cent.

Also, it is a characteristic of the basic chemistry of fuel manufacture that each octane number becomes progressively more difficult to produce. Therefore an informal prognosis of the rate of gasoline improvement over a period such as that up to and including 1960 must lead to the conclusion that octane numbers will increase but at a progressively slower rate than in the past.—*From a paper by D. P. Barnard and R. F. Marschner, Standard Oil Co. of Ind., presented at the recent war engineering annual meeting of the S.A.E.*

Yankee Ingenuity and Engineering

TOO much common everyday engineering, development, and the like, are now called research which should not have the dignity of that term. I like to think of fundamental research as the search for and discovery of new facts and unknown laws of nature. Engineering appears in the picture by taking these fundamentals and applying them to the welfare of man. In this process, Yankee ingenuity is what makes the whole thing click.

I believe it originates from two sources. First, the environment of the early settlers forced them to exercise a tremendous amount of ingenuity or starve. Second, I think it is more or less hereditary. From whatever source it originated, it is certainly one of the bases of our present industrial empire. I would for good measure throw in a possible third source which is the ever-present desire to lighten a man's own burden, possibly a remote cousin to laziness, shaded a little by a desire for praise from his friends.

Yanks and Yankee Ingenuity

On the seven seas and on the shores of the seven seas, American forces are known as "Yanks" regardless of race, creed or color, so Yankee ingenuity has become a national symbol, not tied specifically to the original Yankees. Before our boys get through with the present mess on the far flung fronts, Yankee ingenuity will have an international meaning.

There is a very definite connection between ingenuity and engineering. Engineering development and design along established lines will yield a sound product, but for the new short cuts to save labor and material, for the new stunt which makes the product radically better, for the new scheme which makes possible the heretofore impossible, we must look to Yankee ingenuity. In engineering training plus Yankee ingenuity we have an unbeatable combination.

Ingenuity or Executive Ability?

In industry more attention should be paid to the engineers that show this characteristic. Often a highly valuable man is overlooked because he may not have so-called executive ability and is pigeon-holed in some department for that reason. He should be singled out and given an opportunity so that, working in close association with others, this ingenuity will bear fruit many-fold.

The seeking of this should be one of the first duties of the head of an engineering department as new men are taken into an organization.—*From a talk by W. L. Merrill, General Electric Co., at the annual meeting of the A.S.M.E.*

Applying Electronic Controls

ELECTRONIC devices have been applied to industrial equipment engaged in all-out production to a greater extent than many people realize. During 1943 more than 25 billion kilowatt-hours passed through electronic devices, which is more than ten per cent of all electrical energy generated in the United States by any source of power for any purpose whatsoever.

Equipment users familiar with industrial engineering no longer consider the use of electronic control as involving a daring feat or that they are pioneering. Therefore it is difficult to subscribe to the claim that we are on the verge of an electronics age which will be only deferred for the duration. The fact is that industrial electronics is not a promise for tomorrow; it is really an accomplishment of today.

We should regard electronic control simply as another tool to assist application engineers in meeting the varied demands of industry, not something to supersede or replace well-tried and reliable equipment which is well adapted to the particular work it is doing. Electronic control has assumed its place as a partner with other electric equipment, to be used only where electronic control will do the given job easier, better, at less cost, or indeed make it possible to secure a given result.—*From a paper presented by F. H. Penney, General Electric Co., at the recent annual meeting of the A.S.M.E.*

Severe Sulfuric Acid Corrosion Resisted by Welded Monel Chain

Here's another example of the high resistance of *welded* Monel to corrosion.

It's the record established by welded Monel chain, designed and fabricated by the Youngstown Welding and Engineering Co., and used in mechanical picklers by one of the country's largest producers of alloy steels.

In pickling steel this mill employs the highly corrosive solution of 10% sulphuric acid at 150°–165° F.

When chains of high copper alloy were used, the average life before failure was 2½ months.

Welded Monel chain lasts 7 *times as long* . . . a full year-and-a-half.

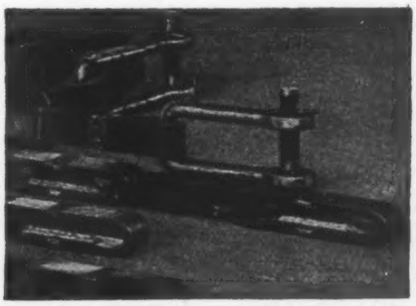
Each of the mechanical picklers handles a load of 5 tons. The welded Monel chain, during its service life, averages a total of 20,000 tons.

Welded Monel sling chains are also used in loading the picklers.

Tough and strong, welded Monel forms strong, long-lived equipment where corrosive conditions exist.

THE INTERNATIONAL NICKEL COMPANY, INC.
67 Wall Street New York 5, N. Y.

Mechanical picklers at work in one of America's large steel mills. Average load on pickler is 5 tons; tubes and bars up to 6" O.D. and 20 feet long are pickled. Chains move 20 feet per minute.



Close-up of Monel shackles on welded Monel chains, designed and fabricated by Youngstown Welding and Engineering Co., Youngstown, Ohio.

Section of welded Monel chain. Round links are oxy-acetylene welded; flat links and spacers are arc welded using 130X Monel electrode.

★ ★ ★ ★ **INCO NICKEL ALLOYS** ★ ★ ★ ★

MONEL • "K" MONEL • "S" MONEL • "R" MONEL • "KR" MONEL • INCONEL • NICKEL • "Z" NICKEL

Sheet... Strip... Rod... Tubing... Wire... Castings

Sheet-Metal Fastenings

(*Concluded from Page* 136)

formulation of intelligent principles for fastener design. A knowledge of these principles will eventually lead to a considerable decrease in the amount of empirical experimentation required in the design of engineering connections. This will be accomplished through the substitution of flexible, analytic design methods in place of the present procedure of static and fatigue testing of an innumerable number of samples until a statistical conclusion is reached. Furthermore, these principles will serve as scientific guides for the future development of improved fastening methods.

Appendix

Based on simplified concepts of joint action, the method of calculating load distribution in multi-fastener joints is essentially geometrical in character. Equalities are written between the relative deflections of the mid-planes of the two sheets and the deflections across the fasteners. It is assumed that the longitudinal sheet strains (averaged across the sheet thickness and in the direction of the joint load) do not vary either between adjacent rows of fasteners or along the joint width. Effects of bending of the sheets are neglected. The implications of these assumptions will not be considered in detail here; it is sufficient to note at the present time that the development presented here is checked experimentally.

Elements of the method can be described with the least amount of complication by considering the load distribution in specific joints. Extension of the work to cover joints with different patterns from those about to be described entails no additional analytical difficulties.

In addition to the physical properties of the sheet material and the joint geometry, the method presumes a knowledge of the curve relating the fastener load to the deflection across the fastener. As is mentioned in the body of the paper, this can be obtained by measurement from tests of two-row joints.

For a three-row joint the notation employed is shown in *Fig.* 11*a*. Fastener deflections are called δ. Deflections of the upper sheet between fasteners are denoted by a; deflections of the lower sheet between fasteners are called a'. Subscripts $_1$, $_2$, $_3$ refer to the three fasteners, in line with the applied load P; double subscripts, such as $_{12}$, $_{23}$, denote sheet portions between the correspondingly numbered fasteners. The fastener loads are denoted by P_1, P_2 and P_3. The uniform fastener spacing is m.

From conditions of symmetry, $\delta_1 = \delta_3$, $a_{12} = a_{23}'$ and $a_{23} = a_{12}'$. Taking into account the simplifying assumptions already mentioned:

$$a_{12}=\frac{(P-P_1)m}{wtE} \qquad (1)$$

where P is the applied load associated with each line of fasteners, t is the sheet thickness, w is the fastener spacing in the direction of the joint width and E is the Young's modulus of the sheet material. Also,

$$a_{23}=\frac{P_1 m}{wtE} \qquad (2)$$

From the geometry of *Fig.* 11*a*, it is evident that

$$\delta_1+a_{12}'=\delta_2+a_{12} \qquad (3)$$

or

$$\delta_1-\delta_2=\frac{(P-2P_1)m}{wtE} \qquad (4)$$

The conditions for static equilibrium require that

$$P=2P_1+P_2 \qquad (5)$$

and so Equation 4 can be rewritten

$$\delta_1-\delta_2=\frac{P_2 m}{wtE} \qquad (6)$$

Since the relationship between the fastener load and the fastener deflection is assumed to be known Equations 5 and 6 are two simultaneous equations involving the two unknowns P_1 and P_2.

A direct method for calculating the load distribution in the joint for all values of the total load P is as follows: Choose a value for P_2 and find the corresponding value of δ_2 from the load-relative deflection curve for the fastener. From Equation 6 calculate δ_1 and then find P_1. From Equation 5 compute the total load P. Repetition of this procedure for several values of P will allow the construction of curves relating the magnitudes of the fastener loads P_1 and P_2 to the total joint load P.

The experimental check of the method requires that P_1 be calculated directly from the data measured on three-row joints. This is done by means of the relation

$$P_1=\frac{a_{23}}{a_{12}+a_{23}}P \qquad (7)$$

Analysis of the four-row joint is similar in outline to the previous solution. The notation used is similar to that defined for the three-row joint and is shown schematically in *Fig.* 11*b*. Resulting equations are:

$$P=2(P_1+P_2) \qquad (8)$$

and

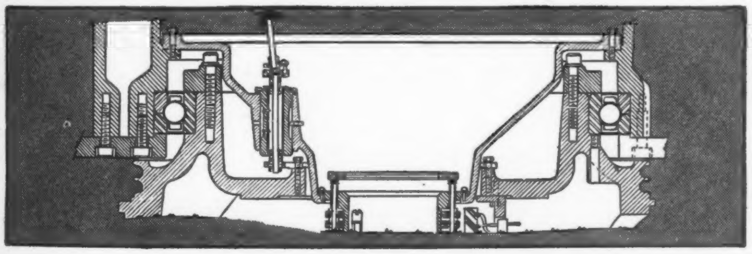
$$a_1-a_2=\frac{2P_2 m}{wtE} \qquad (9)$$

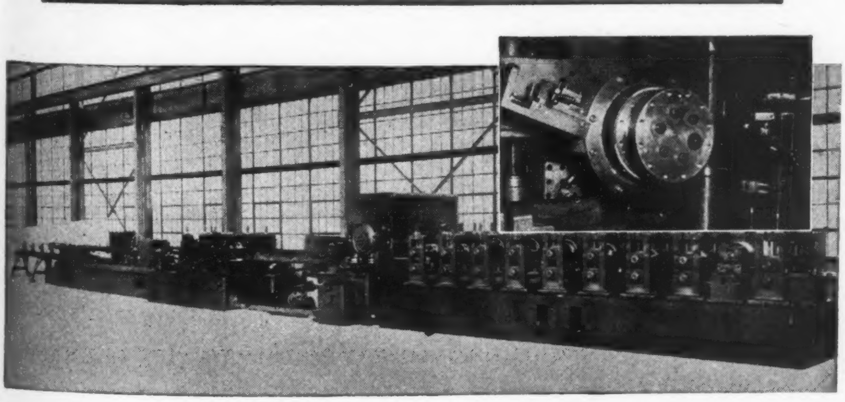
Since the load-relative deflection curve for an individual fastener is presumed to be known, Equations 8 and 9 form two simultaneous equations in the two unknowns P_1 and P_2. A calculation procedure similar to that outlined for the case of the three-row joint can be used to facilitate their solution.

IN THE NEWS

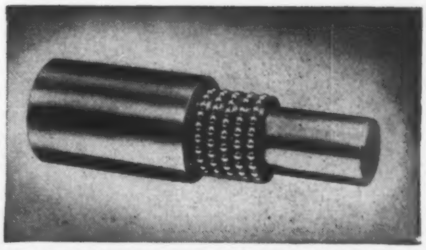
WITH TORRINGTON BEARINGS

SIMULATING FLIGHT CONDITIONS, the Celestial Navigation Trainer, designed and built by Link Aviation Devices, Inc., provides operational flight crew training for navigators, bombardiers, radio operators and pilots. A fuselage which is a reproduction of a bomber nose is mounted on a universal joint and can duplicate the bank, pitch and turn of a plane under normal instrument flight. A large, 23-inch diameter ball radial bearing, supplied by Torrington's Bantam Bearings Division, helps support the fuselage, enables it to rotate with anti-friction ease and smoothness in response to the pilot's touch on controls. Photo above shows "terrain" as seen by navigator or bombardier.

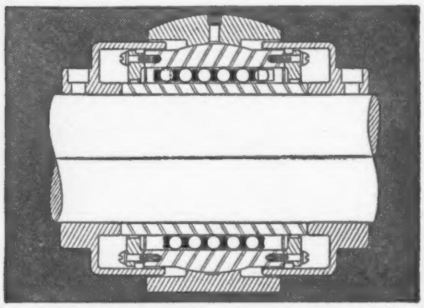




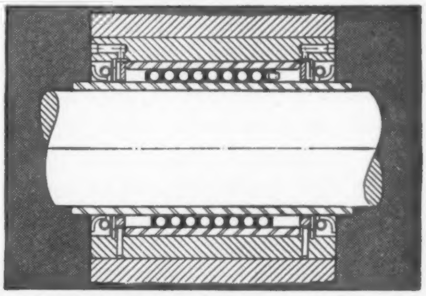
"FROM STRIP TO TUBING" describes the function of this 500 KVA Tube Welding Machine designed and built by The Yoder Company for feeding flat strip steel in, taking finished tubing out. Main electrode support shown in inset, rotates on a special 20" O.D. ball radial bearing supplied by Bantam for this "out-of-the-ordinary" application. Anti-friction bearings in every major type and in sizes up to 10 *feet* in O.D. are supplied by Torrington's Bantam Bearings Division. If you have a difficult or out-of-the-ordinary bearing job, TURN TO TORRINGTON for experienced assistance.



(1) NEW TYPE BALL RECIPROCATING BEARING that offers unusually high capacity is now available. Designed by engineers at the Bantam Bearings Division of The Torrington Company, this new anti-friction unit has a "spirally wound" retaining cage containing a full complement of balls over a long axial distance, and provides more than double the contact than heretofore available in a reciprocating bearing.



(2) NEW APPLICATIONS are expected from this new high capacity unit which offers the design engineer several advantages over the type formerly available. This X-section shows a typical mounting suggested for printing press inking rolls, designed to eliminate possibility of oil leakage.



(3) OTHER APPLICATIONS are expected to be found in engine governors, welding machine guides, doctor blades, coating machines, spool winders, buffers, polishers, etc. A number of methods of mounting are suitable, depending on existing design factors. A second suggested type is shown here.

A FULL RANGE OF NEEDLE BEARINGS to meet virtually any radial load requirement is included in the Torrington-Bantam line, as well as all major types of tapered roller, straight roller and ball bearings. Our engineers will be glad to work with you in selecting the right type for your requirements. Join the TURN TO TORRINGTON for your bearing needs.

ARMY E NAVY

TORRINGTON BEARINGS

STRAIGHT ROLLER · TAPERED ROLLER · NEEDLE · BALL

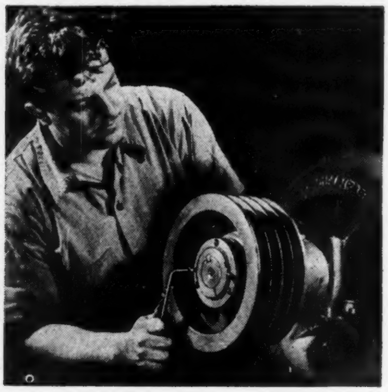
THE TORRINGTON COMPANY · BANTAM BEARINGS DIVISION

SOUTH BEND 21, INDIANA

New PARTS AND MATERIALS

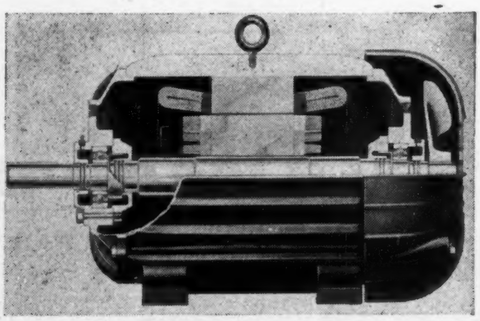
Self-Centering Sheave

ANNOUNCED by Allis-Chalmers Mfg. Co., Milwaukee, is a new sheave known as the "Magic-Grip" sheave which is designed for quick and easy mounting and demounting. It locks to shaft in one tightening operation. As its tapered split bushing, which accommodates normal shaft tolerances, is drawn further into the sheave, the bushing, sheave and shaft are locked together simultaneously. This positive clamp fit insures that the sheave is centered and secure. The new design of the sheave permits it to be mounted closer to motor, increasing bearing life by reducing shaft overhang.



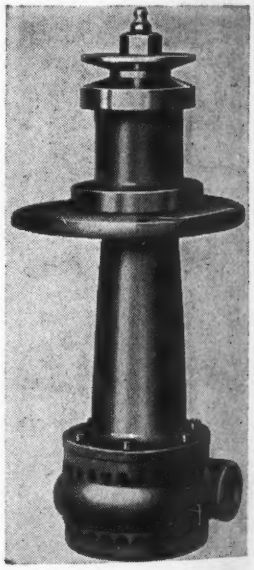
Corrosion-Resistant Motor

SUITABLE FOR operation in atmospheres containing injurious dusts, corrosive vapors or gases, and excessive moisture, such as are often encountered in chemical plants, textile mills, food plants and mines, a corrosion-resistant motor has been made available by The Crocker-Wheeler division of Joshua Hendy Iron Works, Ampere, N. J. The motor is obtainable in sizes from 1 to 15 horsepower, and for operation from any polyphase power supply. While the new unit is of the totally enclosed, fan-cooled type, the design departs from previous models in that there are no cooling ducts to become fouled with wet or sticky dusts. Exposed parts are acid and alkaline resistant. In addition to the mechanical sealing of the entire motor, each coil is individually sealed against moisture, fumes, vapor and dust by the vacuum impregnation process. In one test the motor was operated on an intermittent cycle while drenched in water for 48 hours without failure, and in another a fine chalk was introduced into the test chamber for several hours without penetration into the motor.



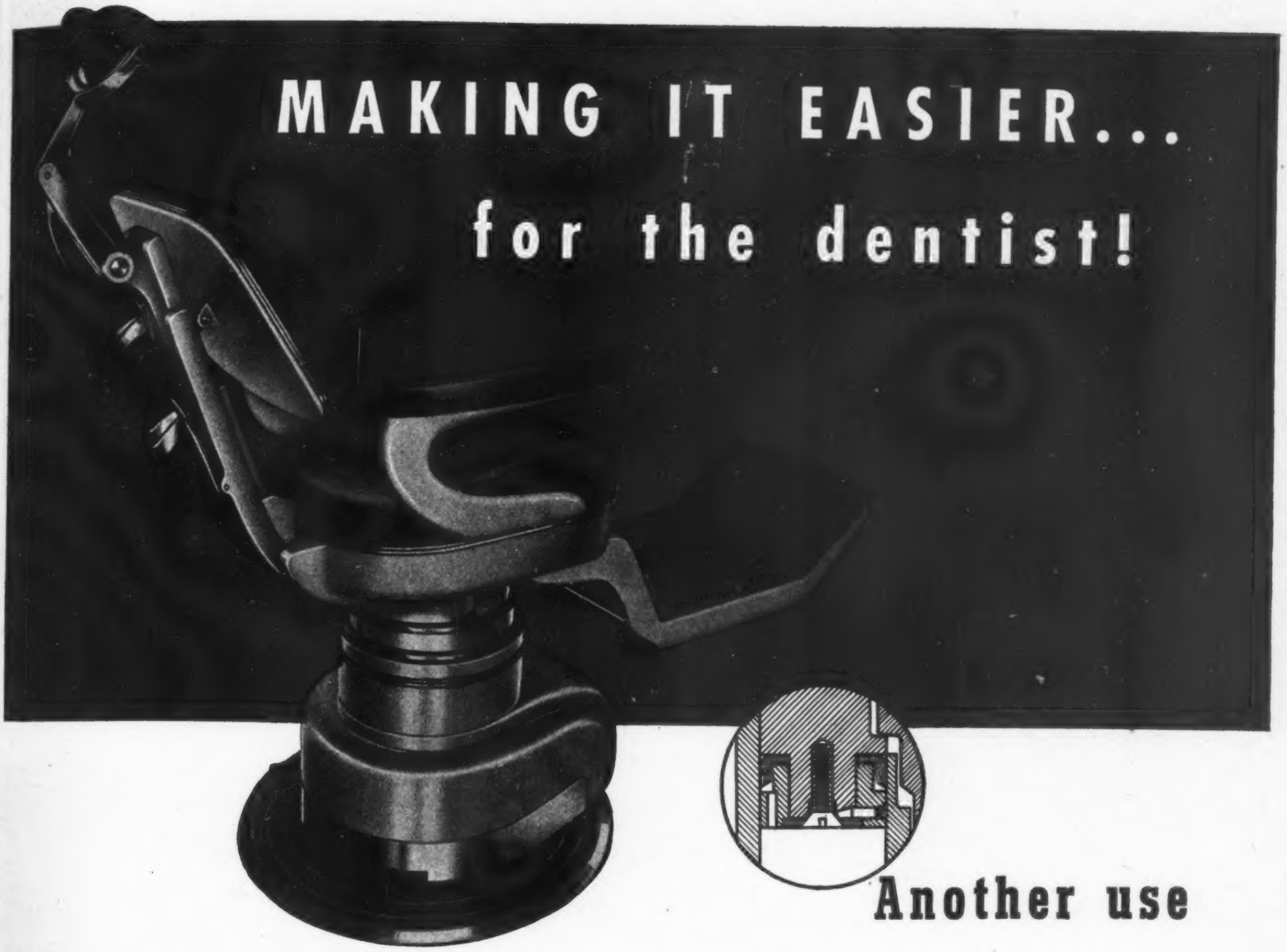
Belt-Driven Coolant Pump

KNOWN AS Model 1-P3, a new belt-driven gusher pump is being placed on the market by The Ruthman Machinery Co., 1818 Reading road, Cincinnati 2. Operated in a vertical position, the pump is propelled by a No. A V-belt using a detached motor or drive-shaft. A one-piece shaft is employed which revolves on ball bearings mounted within the tubular section between the plate flange and the "V" pulley. The pump stem, flange and impeller housing used in the construction of this belt-driven model are interchangeable with, and available in two lengths similar to the motor-driven gusher pump.



Brushes for Wide-Range Operation

DEVELOPED recently by National Carbon Co. Inc., Madison avenue and West 117th street, Cleveland, a series of brushes permits a range of operating characteristics sufficiently broad to cover practically every type of rotating electrical equipment used on aircraft. The brushes are designed to give satisfactory performance under high-altitude, sea level and high-humidity conditions. This affords protection of aircraft electrical equipment during the range of flight and under adverse conditions. The new brushes have in their composition accurately determined amounts of special material, maintaining a protec-



Another use for Vim Leather hydraulic packings . . .

There is no industry or profession in which hydraulics are not used these days—even dentistry.

The patient has his ups and downs by the mere touch of a foot pedal in this S. S. White "Diamond" chair, enabling the dentist to get a better view of uppers or lowers.

It's a smooth ride, because Vim Leather Cup Packings are used on the pump piston and cylinder as a sure seal against leakage or irregular operation at low pressure.

Vim Leather Packings are used everywhere in hydraulic or pneumatic mechanisms. They're engineered to the job—a service design engineers find invaluable.

For that post-war planning, still in the design stage, call in The Houghton Man for data and advice on packing problems.

E. F. HOUGHTON & CO.

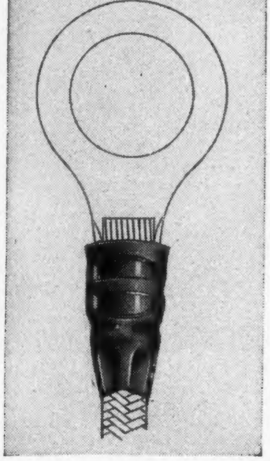
303 W. Lehigh Avenue • Philadelphia 33, Penna.

Sales and Service in All Principal Cities

HOUGHTON'S Engineered VIM Leather Packings

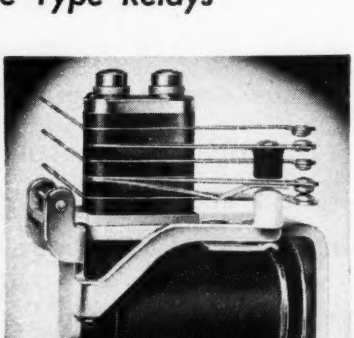
tive film on commutators and slip rings in dry nitrogen and at atmospheric pressure as low as two millimeters of mercury. While the new grades of brushes were primarily developed for operation under conditions of low temperature, low humidity and rarified atmosphere, they have also shown performance at sea level and under conditions of high himidity comparable to that of corresponding grades not designed for high-altitude operation.

Pre-insulated Terminal Designed



PRE-INSULATION of terminals announced by Aircraft-Marine Products Inc., 1591-0 North Fourth street, Harrisburg, Pa., eliminates need for applying insulating sleeving to crimped terminals. The new terminals require only one operation—merely crimping of the terminal on the wire with precision installation dies. Pre-insulation takes the exact contour of the crimp without distortion, cracking, or drying out in use. Identification of terminals and matching dies is made by marking each of the two sizes with a distinctive color—red for terminals and dies for wire sizes 22 to 18, blue for wire sizes 16 to 14. Insulation is permanently bonded to the copper of the terminal so that it cannot be accidently removed. Dry dielectric strength is 750 volts per mil; wet, 350 volts per mil; and tensile strength is 2150 pounds per square inch. Life at 220 degrees Fahr. for the terminal is over 400 hours. The terminal does not shatter when pinched with pliers at minus 40 degrees Fahr.

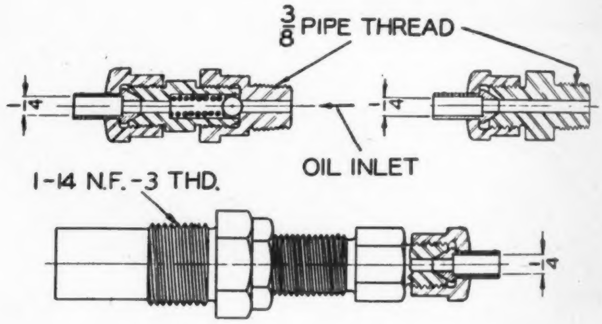
Telephone Type Relays



TELEPHONE type relays announced by Allied Control Co. Inc. 2 East End avenue, New York 21, incorporate newly developed materials and precision construction for dependable life under severe operating conditions. Model TSU, illustrated, for bottom mounting, is one of the smaller units in this new line of relays. Specifically designed for crystal switching, the relay is suitable for various high frequency, plate circuit and general utility applications. Contact arrangements can be supplied normally open, normally closed, single-pole double-throw, or in any two combinations. The contacts will carry two amperes at voltages to 24 volts direct current and 115 volts alternating current, noninductive load. Coil resistances are available from a fraction of an ohm to 5000 ohms. Model TSL for end mounting is similar to the TSU type, both being available with ceramic or bakelite insulation. Overall dimensions are 1¼ inches long, ¾-inch deep and 1½ inches high. Height includes maximum number of contact arrangements. Weights for both are 1¼ ounces. These relays will withstand vibrations up to 10 G and meet all standard salt spray and humidity specifications.

Hydraulic Fittings Announced

HIGH-PRESSURE hydraulic fittings for 3000-pound hydraulic working pressure have been announced by Anker-Holth Mfg. Co., 332 South Michigan avenue, Chicago 4. Pipe connections are ⅜-inch and soldered connections are for ¼-inch outside diameter by .05 wall seamless steel tubing. Standard high-pressure assemblies include: A



check-valve which is reversible by placing the steel ball ahead of the spring; a safety valve, which is spring-loaded and adjustable; a straight connection; tee and elbow assemblies; and ⅜-inch and ½-inch four-way operating valves. Cut from solid steel bar stock, these fittings are designed with a factor of safety of 50 per cent and are hydraulically tested before shipment.

Polyvinyl Resin Group Offered

NOW BEING offered by The B. F. Goodrich Co., Akron, O. to industrial users is a group of polyvinyl resins known as Geon. Two of the resins, Geon 202 and 203 are new vinyl chloride, vinylidene chloride copolymers—different from others developed in this group. The 100 series of the group are special vinyl chloride polymers characterized by their thermal and light stability, toughness and chemical inertness. Type 101 has been developed especially for electrical applications such as wire and cable insulation, while 102 is adaptable to general services. The new 200 series was created to meet the need for polyvinyls which combine increased solubility and thermoplasticity with exceptional stability, chemical resistance and wide temperature range. Their resistance to hydrolysis by boiling water or even hot alkali is outstanding in the field of vinyl chloride polymers, and like the 100 series they have

Small Size and Compact Design are Features of these Needle Bearings

•

REQUIRE LITTLE OR NO MORE SPACE THAN A PLAIN BEARING OF COMPARABLE RATING

•

One of the outstanding advantages of Torrington Needle Bearings is their exceptionally small size in relation to their unusually high radial load carrying capacity.

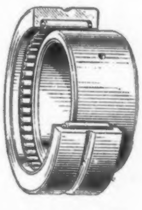
The use of a full complement of small diameter rollers within a single retaining raceway—the basic design principle of all Torrington Needle Bearings—has provided the advantage of *anti-friction operation* within space limits no larger than required for a sleeve or plain bushing.

The small size which characterizes the Needle Bearing also contributes to efficiency and economy of product design in other important ways: it facilitates compact design; simplifies housing structures, frequently permits reduction in overall size; contributes to lighter weight; and through one or more of these advantages—makes for greater efficiency and lower cost.

Size Advantage in Newer Type Needle Bearings

As one new application led to another, modifications were made and new type Needle Bearings were developed by Torrington to meet new engineering requirements which were encountered, particularly in aircraft applications. For example, the AT Type, originally developed for use in conjunction with AN bolts, has a heavy, through-hardened outer race, a heavy hardened inner race, and is designed to give a maximum static non-brinell capacity equal to the sheer strength of the AN bolt. Another type was designed to adapt the Needle Bearing principle to cam-follower service...another for "x-tra capacity" load requirements... and so on until today Torrington provides a Needle Bearing to meet virtually any radial load requirement. The size ranges of the principle types are shown in the accompanying table. Additional information on other types and sizes will be found in our Needle Bearing Catalog, available on request.

TYPE DC



TYPE NCS

TYPE AT

TYPE RC

TYPE NEEDLE BEARING	MINIMUM SHAFT DIAMETER	MAXIMUM SHAFT DIAMETER
DC	.1875"	3.500"
NCS	.6250"	5.500"
AT*	.1900"	1.000"
RC	.2500**	1.500**

*Size range to fit Standard AN Bolts Nos. 3 to 16

**Indicates OD of stud on cam-follower. Camroll diameters range from .6875" to 4.000".

Importance of Size in Postwar Product Design

War-accelerated design of military equipment and material which has shown a definite trend toward "compactness" to save cargo space and lighter weight to facilitate air transport will continue to promote greater efficiency in the design of postwar products. The contributions which Needle Bearings can make in this direction are already well established in many fields. If you are planning new products or design improvements in old, it will pay you to investigate fully the advantages in the use of Needle Bearings. Our engineering department will welcome the opportunity to cooperate in developing designs or assist in the selection of the correct Needle Bearing for your requirements.

THE TORRINGTON COMPANY

Established 1866 • Torrington, Conn. • South Bend 21, Ind.
"Makers of Needle Bearings and Needle Bearing Rollers"

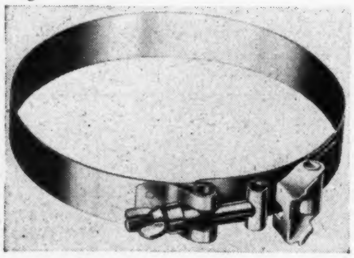
New York • Boston • Philadelphia • Detroit
Cleveland • Seattle • Chicago • San Francisco
Los Angeles • Toronto • London, England

TORRINGTON

NEEDLE BEARINGS

unusual stability to light and heat. These resins when compounded with other materials can be processed in many ways, including injection and compression molding, extruding, calendering, solution coating and film casting. Compositions varying from rigid thermoplastic to very soft jelly can be obtained. In addition to furnishing the resins in powder form with compounding done by the customer, the chemical division of the company will supply them in ready-to-use form, either as granules or sheet. Standard compounds are available for many uses, such as electrical insulation, automotive and aircraft tubing.

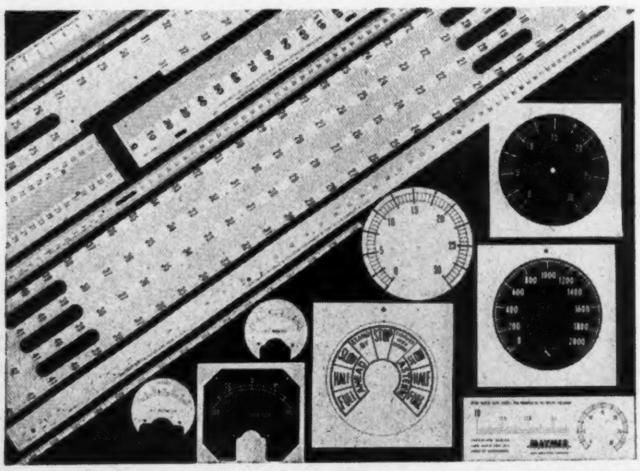
Clamp for Removable Equipment



PRODUCED BY Marman Products Co. Inc., Ingelwood, Calif., a new type clamp known as the "Quick-Coupler" combines a snap-on latch with the full adjustment of a standard clamp, all in one unit. The design permits instantaneous removal of clamp or its quick installation. Available in all sizes in aluminum alloy and stainless steel, and in shapes to fit all convex surfaces, the new clamp may be had with self-locking, plain hex or wing nuts. Like all clamps manufactured by this company, the new one embodies certain features which permit the application of a uniform radial pressure to the periphery of the parts over which it is installed and its rugged construction allows re-use without efficiency loss. The clamp is suited for handling removable equipment of various types.

Scales, Dials, Templets Offered

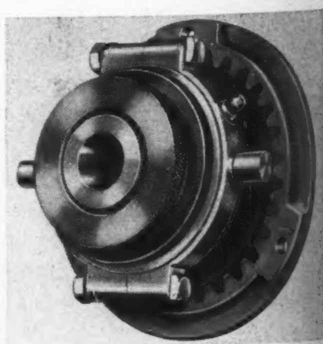
AN EXCLUSIVE process for making scales, dials, templets, indicators, diagrams and other precision units requiring a silhouette of extreme contrast has been perfected by The Maymer Corp., 10880 Berea road, Cleveland 2. According to the company, scales and other similar units are being reproduced from master art work or to customers' specifications on plain or coated flat-surfaced metals, plastics and other stable surfaces. The units have easily read numerals and graduations with total absence of confusing background glare. Continuous scales up to 100 inches in standard lengths—also special sizes and shapes—are being produced. Regardless of length or shape, precision is assured by the "Syl-o-ette" process.



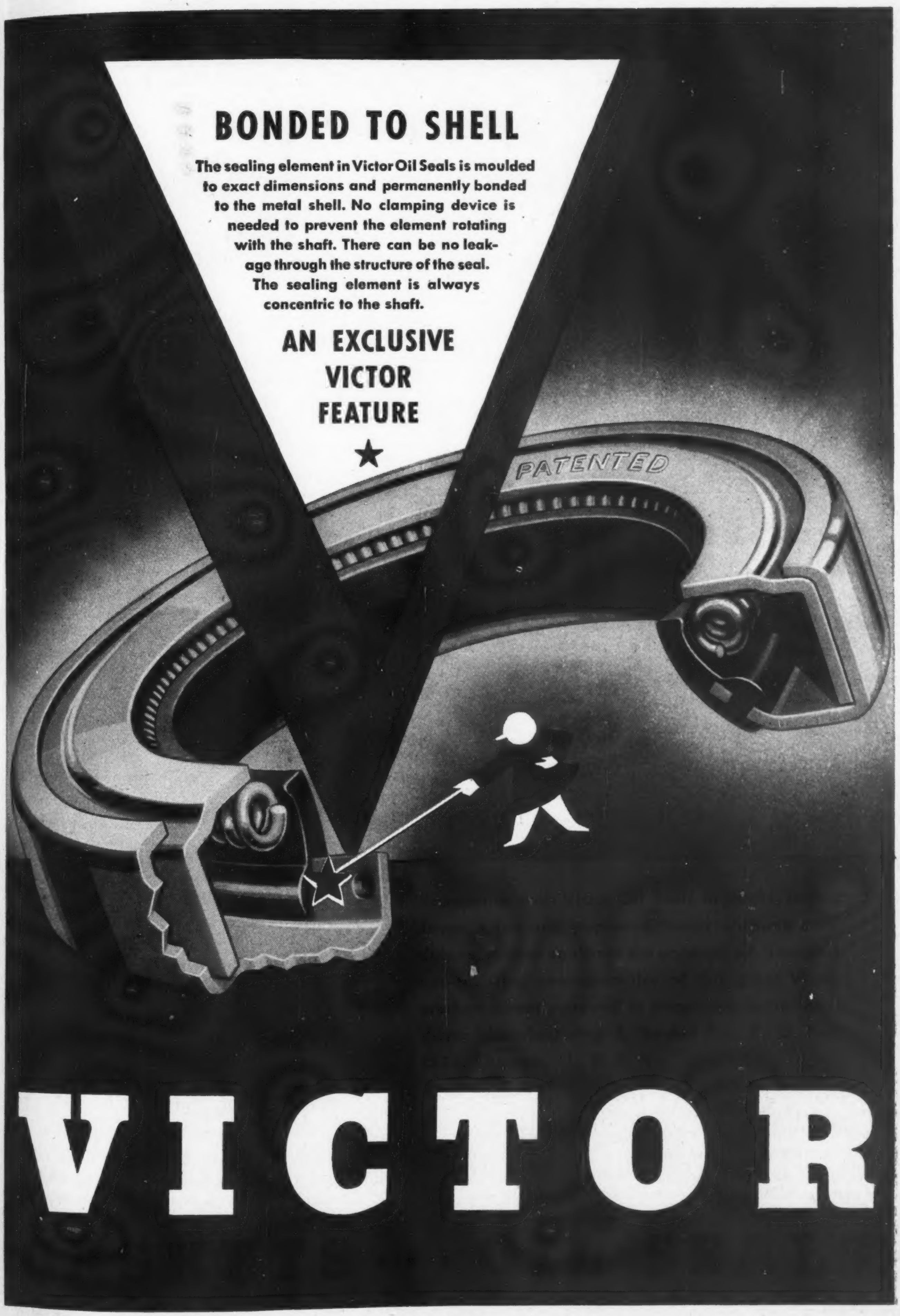
Glass Fabric and Resin Plastic

AVAILABLE in many forms, a new material known as Corresite has been developed by S. Blickman Inc., Weehawken, N. J. A combination of glass fabric and chemically resistant synthetic resin, it is corrosion resistant to a wide variety of reagents and has high mechanical strength. Made in different grades, properties and shapes, it also can be modified to suit various applications. Material has tensile strength of 36,000 pounds per square inch, heat resistance to 270 degrees Fahr. and low electrical and thermal conductivity. Light in weight, its specific gravity is less than one-quarter that of steel. It can be sawed, drilled, turned on a lathe, threaded, and, in general, can be machined like metal. Several grades of the material have been developed for applications such as pipes, hoods, linings, troughs, cylinders and other process equipment. Because of its high strength, it is also suitable for structural parts subject to corrosive conditions. Pipe and tubing available in this material are furnished in diameters ½-inch and larger and in wall thicknesses from ⅛-inch up. A complete line of Corresite fittings, including bends, tees, crosses and couplings are also furnished for use with pipe made of this material

Rolling-Grip Friction Clutch



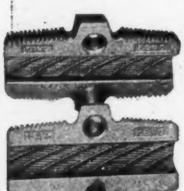
APPLICABLE to packaging, floor scrubbing, paper handling, bread wrapping, canning, fruit grading, textile and many other types of machines, the new rolling - grip friction clutch of Dodge Mfg. Corp., Mishawaka, Ind., is a compact, rugged and easy-to-operate disk clutch for use where power requirements are relatively small. It is available with a 3½-inch diameter disk, rated at ⅓ horsepower at 100 revolutions per minute, or with a 4½-inch diameter disk, rated at one horsepower at 100 revolutions per minute. There are no toggles; instead a number of highly polished hardened steel balls are forced into a V-shaped groove by a hardened steel cam. Forcing the balls toward the center of the shaft widens the V-shaped groove. This application of the basic principle of the inclined plane or wedge produces a heavy power-transmitting pressure on the friction surface. There are no

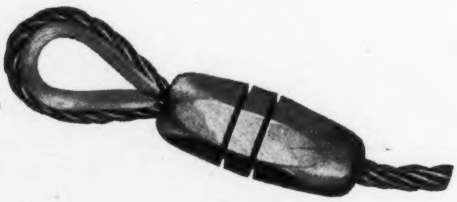

BONDED TO SHELL
The sealing element in Victor Oil Seals is moulded to exact dimensions and permanently bonded to the metal shell. No clamping device is needed to prevent the element rotating with the shaft. There can be no leakage through the structure of the seal. The sealing element is always concentric to the shaft.
AN EXCLUSIVE VICTOR FEATURE
PATENTED
VICTOR

links, pins, cotters, or other highly stressed mechanical parts to wear or break. Adjustment is controlled by threaded collar locked by setscrews. The clutch may be adapted to any required operating conditions. For example, it may be provided with a gear tooth friction disk and drive ring or with bolted friction disk, either being suitable for operation under dry conditions. To operate in oil, asbestos-fiber friction disks may be replaced with thin metal disks. The cam and thrust ring contours may be altered to change clutch characteristics.

Wire Rope Clamp in All Sizes

HERETOFORE available in nine sizes, clamps for wire ropes are now being furnished in fifteen—from 1/16 to ¾-inch—by National Production Co., 5007 St. Jean avenue, Detroit 13. The clamp is made in two halves, each made to fit the rope. The large grooves pocket the large spiral strands and hold the rope from endwise slippage. The small grooves pocket each small wire that makes up the strands and prevent the rope from spiral winding out of the clamp. All strands and surface wire contacts the full length inner surface of the clamp. Both halves are gripped tightly on the rope with strong alloy steel nuts. Tested and approved for use on high tensile strength wire rope, the clamp is free from projecting bolts and nuts, and sharp ends of wires are covered. Having a surface that is rounded and smooth, the clamp will not catch or foul on any mechanical apparatus with which it may come in contact. Of high tensile strength alloy steel, and coined to accurate dimensions, the clamp has a great holding power and maintains its grip beyond the breaking strength of the wire itself.





Bevel Gear Universal Joints

APPLICABLE FOR remote control of valves, pumps, engines, ventilators, turrets and other equipment in inaccessible locations, or for centralized control of widely separated units, Condenser Service & Engineering Co. Inc., 95 River St., Hoboken, N. J., has announced its bevel gear universal joints. This new type of manganese bronze universal joint provides a simple, flexible solution to the problem of shaft line-up. They are designed and built to give positive transmission of torsion regardless of shaft angle, operating equally well at from 0 to 135 degrees on vertical center line, or from 0 to 360 degrees on horizontal center line. This angular range is covered without additional or special gears through use of standard brackets available. Capacity is from 1580 to 4938 inch-pounds at 49 revolutions per minute. Design of gears is such that they cannot ride over the top. To withstand severe service, housings are cast brass. Shafts are CR steel, pressed into the bronze gears and keyed. The joints are available in five standard sizes for shaft diameters of ½, ¾, 1, 1¼ and 1½ inches. Special sizes are also furnished to suit individual requirements.

Fixed and Repeat Cycle Timer

AVAILABILITY of a complete line of continuous running repeat cycle timers (Series 5800) has been announced by Haydon Mfg. Co., Box 52, Forestville, Conn. For use with either alternating or direct-current operation, the timers are furnished with from one to eight switches, as required. Motor speed is selected according to the timing cycles desired; however, most popular speeds are from 8 revolutions per minute down to one revolution per month. Single or double-throw, 10-ampere capacity at 110 volts alternating-current precision snap switches are used. Another assembly (No. 5800-4, four-circuit, alternating-current motor-driven unit) has adjustable cams which can be set to throw the switch from one degree to 180 degrees of the revolution. Specifically cut cams are also available for various combinations of "on" and "off" cycles covering a range of from more than one actuation per second down to one per month. These timers are furnished for various commercial alternating-current voltages and frequencies as well as on direct current, the wide range of speeds being secured through sealed-in, lubricated gear trains. They are designed to meet any desired pattern of electrical impulses in any overall interval for radio keying, industrial timing, process control, sequence switching, etc. They are also available with brake for instant stop for end of single cycle if desired.

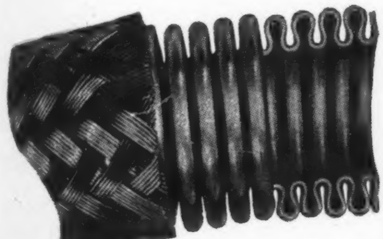
Compensating Finger Chuck

ONE OF the new two-jaw compensating finger chucks offered by Anker-Holth Mfg. Co., 332 South Michigan avenue, Chicago 4, is shown in the accompanying illustration. It is designed for casting shown at the left. This design is arranged so that the work is located from a finished round shoulder on the bottom of the casting and by two hardened and ground pins, which also serve as drivers. Plate on the face of the chuck is hardened and

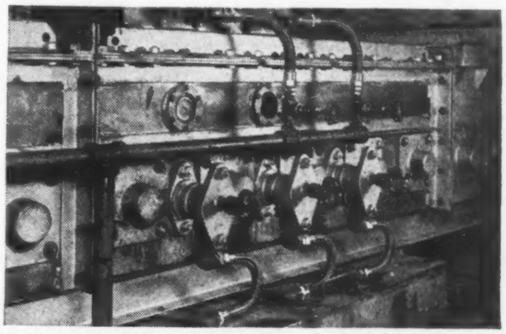
"Flexible as rope STRONG AS PIPE Essential in hundreds of services"

That might be one way to sum up American Metal Hose... the all metal conductor that conveys gases, oil, steam, water, air, solvents, chips and dust under the widest range of temperatures, pressures and exposures to be encountered in industry... the flexible tubing that isolates vibration, connects misaligned parts, loads and discharges cargoes... provides rigid pipe security in connecting moving machine parts with gas, oil, air, water, or steam supply.

AMERICAN SEAMLESS

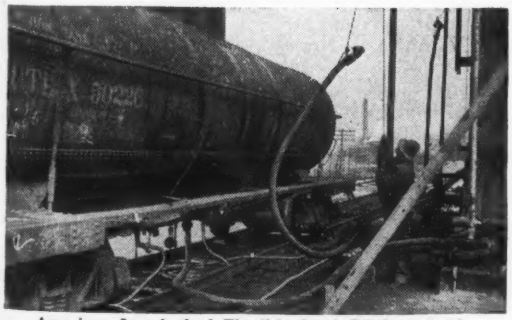


American Seamless. For applications where 100% tightness is essential. Made from *seamless* metal tubes—no packing, joints, welds or laps—the best "insurance" against leakage. Ideal for conveying air, oil, steam, water, volatile liquids and gases. Wire braid "jackets" reinforce tubing for high pressure services. Available in any workable metal.



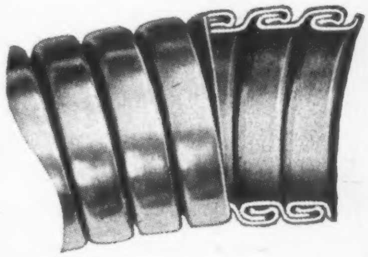
American Seamless Flexible Bronze Tubing insures tight, free-flowing water connections to safeguard the accurate, continuous performance of this shell-case annealing furnace.

AMERICAN INTERLOCKED

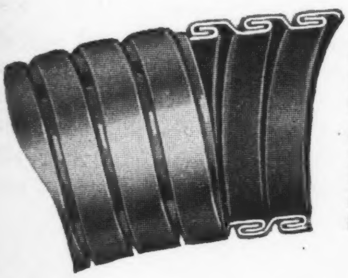


American Interlocked Flexible Steel Oil hose loads and discharges tank cars, may be used on practically any cargo that will flow through a pipe.

American Interlocked. Strip metal, wound into "full-interlock", packed with asbestos, forms the toughest flexible hose for all-round service... the strength and last-ability of metal plus extreme flexibility for ease in handling. Made in bronze for steam and water—galvanized steel for oils...also in other workable metals.



AMERICAN TYPE U.I.



American Type U.I. Full interlocked, but unpacked... a lightweight flexible metal hose for carrying hot air, grinder dust, chips or light dry material. Made from brass, galvanized, stainless steel, aluminum, in sizes to 8" I.D.

44193

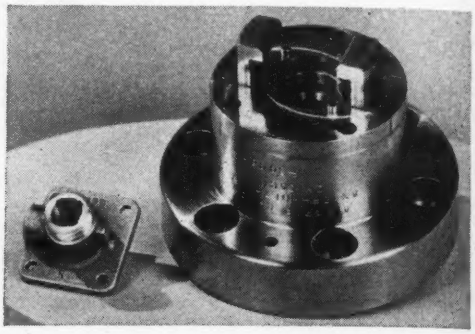
American Flexible Metal Hose, Type U.I., fights the boll weevil in conveying chemical powder to the spray heads on this Cotton Dusting Machine.

ANACONDA

American Metal Hose

AMERICAN METAL HOSE BRANCH OF THE AMERICAN BRASS COMPANY • General Offices: Waterbury 88, Conn.
Subsidiary of Anaconda Copper Mining Company • *In Canada:* ANACONDA AMERICAN BRASS LTD., *New Toronto, Ontario*

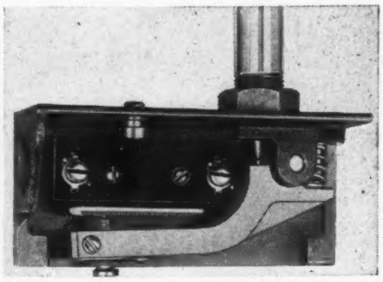
ground. The two jaws are designed so that either one may pull in farther than the other to compensate for varying thickness of rough surface work pieces. Chuck body is made for direct mounting on any standard spindle nose,



but it can be mounted on any spindle by using an adaptor. Available in various sizes, the chucks are furnished for second operation work which must be concentric with the locating surface within extreme close limits.

Temperature Limit Switch

TEMPERATURE control limit switch known as Model H, recently introduced by The Burling Instrument Co., 253 Springfield avenue, Newark, N. J., is available in normally open, normally closed and single-pole double-throw types. In normally closed position the switch cuts off heat, stops fans and closes valves. Normally open it is used for lighting lamps or ringing bells, while single-pole

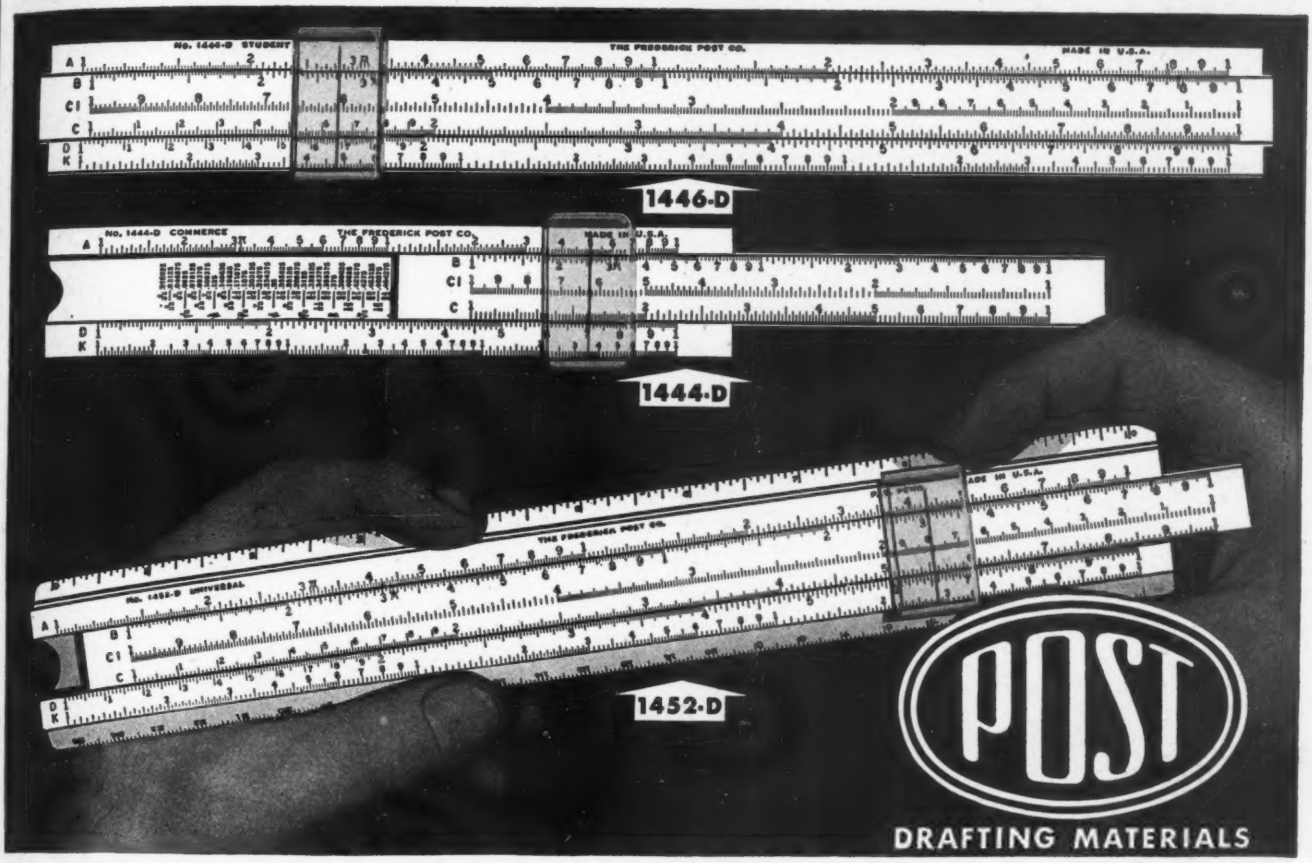


double-throw type breaks heating circuit while closing alarm circuit. Reset button operates from outside of case. Other important features of this type of switch are: Corrosion and heat-resisting tube, dial pointer for easy setting, locking screw for locking temperature setting, terminal plate with large screw terminals, and snap-action switch for eliminating contact troubles. Range of temperature is from 0 to 1400 degrees Fahr.; adjustable range, 200 to 300 degrees Fahr. Dimensions are 5 1/18 x 1¾ x 3 inches.

Ammonia Developing Papers

UNDER THE BRAND name of Casco-Azo a new line of ammonia developing printing papers in blue line, maroon line and sepia transparent sensitized tracing paper is being announced by the John R. Cassell Co., 110 West Forty-second street, New York.

New Slide Rules IN PRODUCTION



3 Types ★ 3 Prices ★ Immediate Delivery

SIMPLE INSTRUCTION MANUALS INCLUDED

Made for the emergency—they'll do the job—and you can get 'em right now. Post Slide Rules come in leatherette cases and are accompanied by comprehensible instruction manuals. Read the descriptions below and reach for the phone for fast-action delivery.

1452-D Ten inch Celluloid Face bevel edge slide rule. Inches and metric rules on bevel edges. Scales A, B, C1, C, D and K on slide front. Scales S, L and T on reverse side.
Attractive leatherette case....................List Price $3.50

1444-D Five inch Celluloid Face POCKET style slide rule. Scales A, B, C1, C, D and K on slide front. Scales S, L and T on reverse.
Leatherette Pocket case.......................List Price $2.25

1446-D Ten Inch Printed Surface slide rule. Inches and metric rules on vertical sides. Scales A, B, C1, C, D and K on slide front. Scales S, L and T on reverse side. Leatherette case..........List Price $1.50

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Buffalo	CLEVE. 0370	New Orleans	RAY. 0331
Chicago	KEY. 7000	New York	WIS. 7-7678
Cincinnati	MAIN 2664	Oklahoma City	3-6306
Cleveland	CHE. 7347	Omaha	ATLANTIC 7890
Columbus	MAIN 3420	Philadelphia	LOM. 7044
Dallas	RIVERSIDE 4403	Pittsburgh	ATL. 3350
Dayton	ADAMS 9174	Portland	ATWATER 8681
Denver	MAIN 5161	St. Louis	CHESTNUT 0688
Detroit	RAN. 8483	Salt Lake City	4-7823
Ft. Wayne	ANT'Y 4142	San Diego	FRANKLIN 1344
Fort Worth	3-3244	San Francisco	DOU. 5975
Hollywd	GRANITE 4188	Seattle	MAIN 4022
Houston	CAPITOL 1233	Tampa	M-8377
Indianapolis	MKT. 4466	Toledo	ADAMS 7224
Jacksonville	5-2166	Tulsa	3-0168
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Forecasting Private Plane

(*Concluded from Page* 126)

on its weight since the heavier it is, the more power is required to operate it. Power is expensive. As shown in *Fig.* 3, the airplane manufacturer is beset, in attempting to control weights, with approximately one-third of the weight-empty of the airplane beyond his control in purchased items. Also, the future's aircraft will have to be more comfortable, smoother and less noisy, and this will involve additional weight over what has been used heretofore.

Factors Influencing Dependability

Although on the whole our prewar designs have proved themselves capable of taking a beating under grueling operating conditions, there have been maintenance troubles on certain petty items which are very disconcerting to the owner. Since the complete aircraft is a complex structure, there is a possibility of an infinite number of things going wrong, and a service test program, similar to the automobile industry's proving ground, will be instigated to catch as many of these items as possible before the aircraft is put into the public's hands.

We know that the basic structure of our prewar aircraft has been satisfactory; only rarely has a structural failure occurred. The main troubles came from so-called accessory items. Experience gained by the war in developing such accessory items to withstand the extreme conditions of wartime use, will be of immeasurable benefit in eliminating most of the difficulties in the postwar period.

There is room for considerable improvement in comfort. Thorough studies of exhaust muffling, vibration, sound-proofing, air conditioning, and seat construction will be made so that the postwar private airplane will be as comfortable and as pleasant to ride in as an automobile.

Comfort Versus Size

Mainly, the difficulty in obtaining comfort in an aircraft is due to insufficient room. Since performanace of an airplane is a function of its size, the designer is reluctant to make the aircraft any larger than the bare minimum; if he does so he cannot meet the performance of his competitors. Although aircraft with sufficient room have been built, their performance has been so impaired that they have not sold as well as their less comfortable but faster contemporaries. A possible solution to this is the use of more efficient, lighter, but higher powered engines, so that extra size aircraft can be dragged through the air at a reasonable speed and cost. Until such power plants are available, however, performance will be partially compromised for comfort.

Due to the uncertainty of the future I do not propose any particular design at this time, but I am sure that super-safe, dependable, comfortable, and inexpensive airplanes with utility increased over prewar models, will be used in considerable numbers by the average individual for pleasure, business, and sport.

Choosing the Right Material

(*Continued from Page* 119)

bearing before starting, the unit load is held down to the order of 250-300 pounds per square inch projected area. The straight copper-lead alloys will stand some ten times this load when operating at 300 degrees Fahr. so the use of alloys of very high compressive strength may not be as necessary as it often is considered to be. Carrying this thought over to the less highly leaded alloys for bushings, it may not be necessary to strengthen the matrix; i.e., 90 copper and 10 per cent lead with no tin might serve in some instances where 80-10-10 has previously been used, since it withstands seizure better than 80-10-10. Another alloy of as good seizure resistance as 80-10-10 and much the same order of compressive strength is 85 copper, 10 lead, 3 tin and 2 per cent antimony, in which antimony replaces several times its weight of tin. Reasonable amounts of zinc appear compatible with this, so scrap from which some of the zinc has been removed could be used to supply both the tin and the copper.

Still another alloy similar in general properties to 80-10-10 is 80 copper, 2½ tin, 2½ zinc, 10 lead and 5 per cent manganese, made for example, from scrap 85-5-5-5 and scrap copper, with the addition of the manganese and of such additional lead beyond that in the scrap as is needed to bring it up to the 10 per cent level.

In fact, the most important feature of a bushing alloy seems to be the 10 per cent lead. Whatever level of strength and hardness the service demands of the matrix—between that of straight copper in 90 copper, 10 lead and that of a bronze with copper and tin in the 80:10 ratio of the 80-10-10 alloy—can be provided from scrap and from hardening elements other than tin.

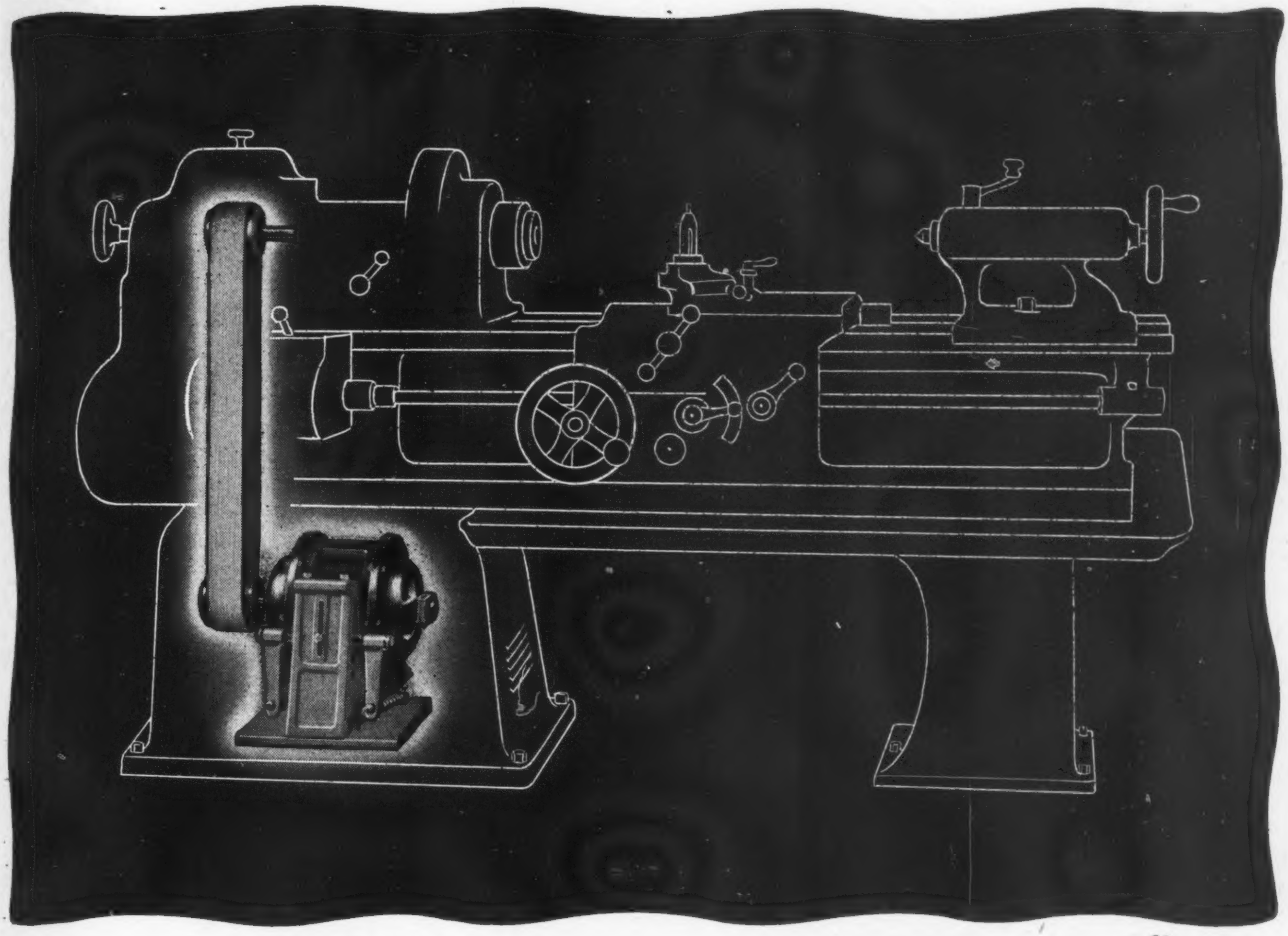
The designer can well make an examination of the needs imposed by the type of service and select an adequate, available alloy that can be made from scrap without use of tin, rather than call for a particular chemical composition that imposes a drain on strategic materials.

Cast Iron Bearings Have Place

Finally, it should be recalled that cast iron with its smearable graphite flakes for initial protection and the tiny oil reservoirs left by these flakes, as well as the porous bushings of common bearing alloy composition or iron, made by powder metallurgy methods and soaked with oil, have their places for some types of service.

It takes only cursory consideration of bearing problems to appreciate the fact that the properties of metals usually measured and specified in the hope of securing quality do not appraise those properties for use. In order to meet the service requirements an assembly, varying from a bearing support plus a solid sleeve bearing to one plus a bearing made up of several layers, has to be engineered out of different materials.

From this example, one may well advance to an equal degree of skepticism about the assumption that the usually measured properties really evaluate the qualities actually involved in other structures and other applications. Measuring unused attributes and overlooking those that



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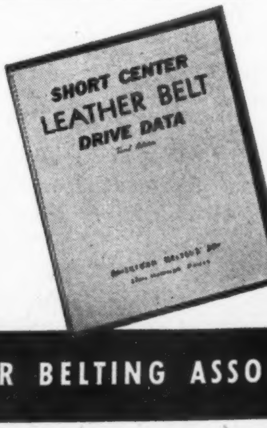

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are used is obviously poor engineering in selection of bearing metals. Other cases are less obvious but often are not much more defensible when the actual demands of service are listed.

Special Simulated Service Testing

Simulated-service tests of the particular part being studied are much cheaper to operate and give more direct and reliable results than do service tests outside the laboratory in which control of variables is less precise. Obtaining correct records is more difficult. Trouble with some part not being studied is likely to delay and to obscure the results. For instance, when trying to test an engine bearing to destruction in an actual engine, it is common to find that failure of some other part occurs often enough to make that kind of approach very time consuming.

Dissecting the machine down to the part to be studied, rigging up to duplicate service loading on the service conditions of that part and running it under close observation greatly simplifies the testing task. Sometimes a scale model will do as well if account is taken of metallurgical size effects.

In such cases as wear testing or corrosion testing, a simulated-service test method must be checked for consistency of results in different tests on specimens of the same uniform material. Until it has been redesigned and adjusted to give consistent results on one material, it is of no use in differentiating among materials. The next step is to use it on a series of three or more materials whose relative behavior in that type of service is known or is obvious. The equipment and method of operation must be such that these are put in their proper order. In that case, the equipment will probably be useful in appraising new materials. At least, it should exclude from further consideration those that are definitely unfitted to withstand the conditions. Since the engineer is not always wise enough to have spotted all the important conditions of service, it remains to corroborate the test indications by actual service.

Gears offer a fine example of the value of the simulated-service method of approach. The early engineering approach to the metallurgy of gear steels assumed that the primary cause of failure was impact and great stress was laid on obtaining toughness of the core of the carburized piece.

Almen and Boegehold[4] have recounted how, by taking a rear axle and putting it through its paces in the laboratory, it was successfully shown that repeated stress rather than impact was responsible, yet that steels did not necessarily behave in gears as behavior in polished fatigue or even in notched fatigue tests would imply. They finally came to the conclusion that the stresses, largely governed by the rigidity of the assembly, i.e., the tooth contact, counted most and that when the assembly was engineered with these factors in view, any one of many steel compositions could be used. Distortion in heat treatment was considered the only important metallurgical factor involved. Incidentally, this was before the days of modern hardenability testing. The indication now is that steels

[4] Almen, J. O., and A. L. Boegehold—"Rear Axle Gears, Factors Which Influence Their Life", *Proceedings, A.S.T.M.*, Vol. 35, Part II, 1935, Pages 99-146.

of equal and complete hardenability should equally resist distortion during heat treatment.

Some of the comment in the discussion of the foregoing paper may well be quoted for their application to present day problems of steel substitution. "There is no evidence that one alloy steel is better than another for rear axle gear application"; "the notched bar impact test has very little engineering value"; "mechanical conditions have a dominant effect on gear life. That there is very little difference in the alloys from which gears are made can be deduced without tests from the fact that many alloys and many heat treatments are in everyday use in production, all presumably giving satisfactory service"; "it is probable that in the majority of machine elements the alloy used is relatively unimportant as compared to mechanical conditions producing stress concentrations, and cheaper steels can be used for many parts now made from the more expensive alloys"; "the fatigue program properly carried out leads to a solution whereas the impact testing would be meaningless, or worse, in that it could give misleading results".

High Alloys Are Not Always Necessary

Almen[5] has recently reiterated the finding that "industry has paid premium prices for alloy steels because of their fancied advantages when used as gears and in other parts. Fatigue tests on actual machine parts correlated with service records have shown that there is no detectable difference between the high-priced alloy steels and many of the low-priced alloy steels when used in many machine elements." He goes on to say that any real differences in fatigue characteristics are "so small in comparison with the mechanical fatigue hazards introduced by the design and fabrication of parts as to be negligible".

The famous "four-square" running-to-destruction test for full-scale rear axles employed in the Chrysler laboratory has led to conclusions about rigidity, etc., corroborating Almen's findings, and emphasizing the importance of tooth contact and the tooth finish.

A case where traditional laboratory tests for mechanical properties and foundry tests for behavior in casting indicated a "candidate substitute" to deserve election but where the indications were incorrect, is cited by Young and Hanink[6]. The question was whether the nickel content of aluminum alloy AMS 4220 for cast cylinder heads could be substituted by iron. As they put it, the laboratory and foundry tests said "yes" but the engine said "no", for every head cracked on experimental running. This does not mean that a test might not be devised that would differentiate without the test of actual service; it means that some property was involved not measured by the conventional tests that were employed. However, it was doubtless simpler, as well as more convincing, to make the engine test than to cast about for some other test that would measure the pertinent property. You are not sure, in a case like this, whether you have selected the right test conditions to correlate with service, without a final service test.

(Continued in next issue)

[5] Almen, J. O.—"Fatigue of Metals as Influenced by Design and Internal Stresses", *Automotive and Aviation Industries,* Vol. 88, Feb., 1943, Pages 28-32, 84.

[6] Young, N., and H. H. Hanink—"New Materials for Aircraft Engines", *S.A.E. Journal,* Vol. 51, May, 1943, Pages 157-164.

Redesigning for Stamping

(*Concluded from Page 123*)

Another use for these low-cost dies is in preliminary development work on high-production parts, enabling the "bugs" to be ironed out of a new design and changes made before the design is frozen into expensive permanent dies. Elsewhere in this issue the use of low-run dies of cast plastics for similar purposes is discussed.

Supplementing the blanking or punching (*Fig.* 8) and drawing (*Fig.* 9) operations, ironing (*Fig.* 15) and extruding (*Fig.* 16) as well as coining often serve to complete the part, insuring the highest degree of accuracy. In the ironing process, *Fig.* 15, a previously drawn cup or shell is reduced in thickness by pushing through tight dies.

Extrusion, *Fig.* 16, places the material of the slug or tube in compression, forcing it to flow through the space between the punch and die. In forming a bushing sleeve for a land mine, SAE 1010 strip steel is rolled into tubular form and then extruded, the rolled tube being reduced in thickness from .14-inch to .124-inch, while the bore remains the same at .501-inch. The finished part is accurate on the diameters to within .0005-inch.

Providing Accuracy without Machining

Sizing and coining operations entirely eliminate machining on certain parts, providing accuracy equivalent to ordinary machine operations. The built-up hub shown at the right of *Fig.* 17 is a case in point. Replacing the machined steel forging at the left, the copper-brazed assembly is sized so as to control all important dimensions. The slight offset on the outside flange is coined to size.

During the period between the two world wars tremendous advances were made in the development and application of stampings, particularly for automobiles, household appliances, and other machines built on a mass-production basis. Thus the most obvious applications, and those affording the greatest opportunities for savings, have been extensively explored. The fresh impetus which the present war has given to the redesign of parts to utilize stampings has greatly broadened the field of application and stampings producers are on the alert for new jobs to tackle.

To provide assistance to designers and engineers, over a hundred leading stamping manufacturers recently formed the Pressed Metal Institute. Special studies are being made of war conversions to determine how the same ideas and techniques can be applied to postwar civilian goods. Aid in solving design problems and in finding sources of supply is offered by the Institute as a service to potential users of stampings.

MACHINE DESIGN is pleased to acknowledge the cooperation of the following organizations in the preparation of this article: Cleveland Ordnance District (*Figs.* 12 and 13); The Commercial Shearing & Stamping Co. (*Fig.* 10); Dayton Rogers Manufacturing Co. (*Fig.* 14); Otto Konigslow Manufacturing Co. (*Fig.* 18); The Midland Steel Products Co.; Morrison Products Inc.; Youngstown Pressed Steel division of Mullins Manufacturing Corp. (*Fig.* 17); National Formetal Co.; Pressed Metal Institute (*Fig.* 7); Pressed Steel Tank Co., (*Fig.* 11); and Worcester Pressed Steel Co.

BUSINESS AND SALES BRIEFS

RESIGNING as assistant to the president of Clearing Machine Corp., Chicago, H. M. Comstock has been named engineering sales representative for the southern California territory at Kropp Forge Co. and Kropp Forge Aviation Co. Mr. Comstock previously had been director of sales at Simplex Products Corp., Cleveland, for more than fifteen years.

♦

Election of Russell M. Allen as vice president in charge of sales has been announced by Allegheny Ludlum Steel Corp. Formerly Mr. Allen had been general manager of sales.

♦

Worthington Pump & Machinery Corp., Harrison, N. J., has announced the appointment of William O. Wilson as a commercial vice president. With headquarters at the Chicago district office, 400 West Madison street, Mr. Wilson will supervise and direct all activities in the Chicago, St. Louis, Kansas City and St. Paul district office territories.

♦

According to a recent announcement The Lincoln Electric Co. has appointed R. H. Davies as welding engineering representative in Washington, D. C. Mr. Davies, who has had wide experience in the industrial engineering field, will be located at 410 Hill building.

♦

Succeeding Robert R. Hirsch, who has resigned, is Walter C. Ahlers as general sales manager of SKF Industries Inc., Philadelphia. For a number of years Mr. Ahlers has been assistant district manager of the Detroit office.

♦

Oliver E. Mount of American Steel Foundries, Chicago, has been elected president of Steel Founders' Society of America.

♦

With M. E. Teague and W. F. Haverstock in charge, Michigan Tool Co. has opened a factory sales district office at 1506 Toledo Trust building, Toledo 4, O.

♦

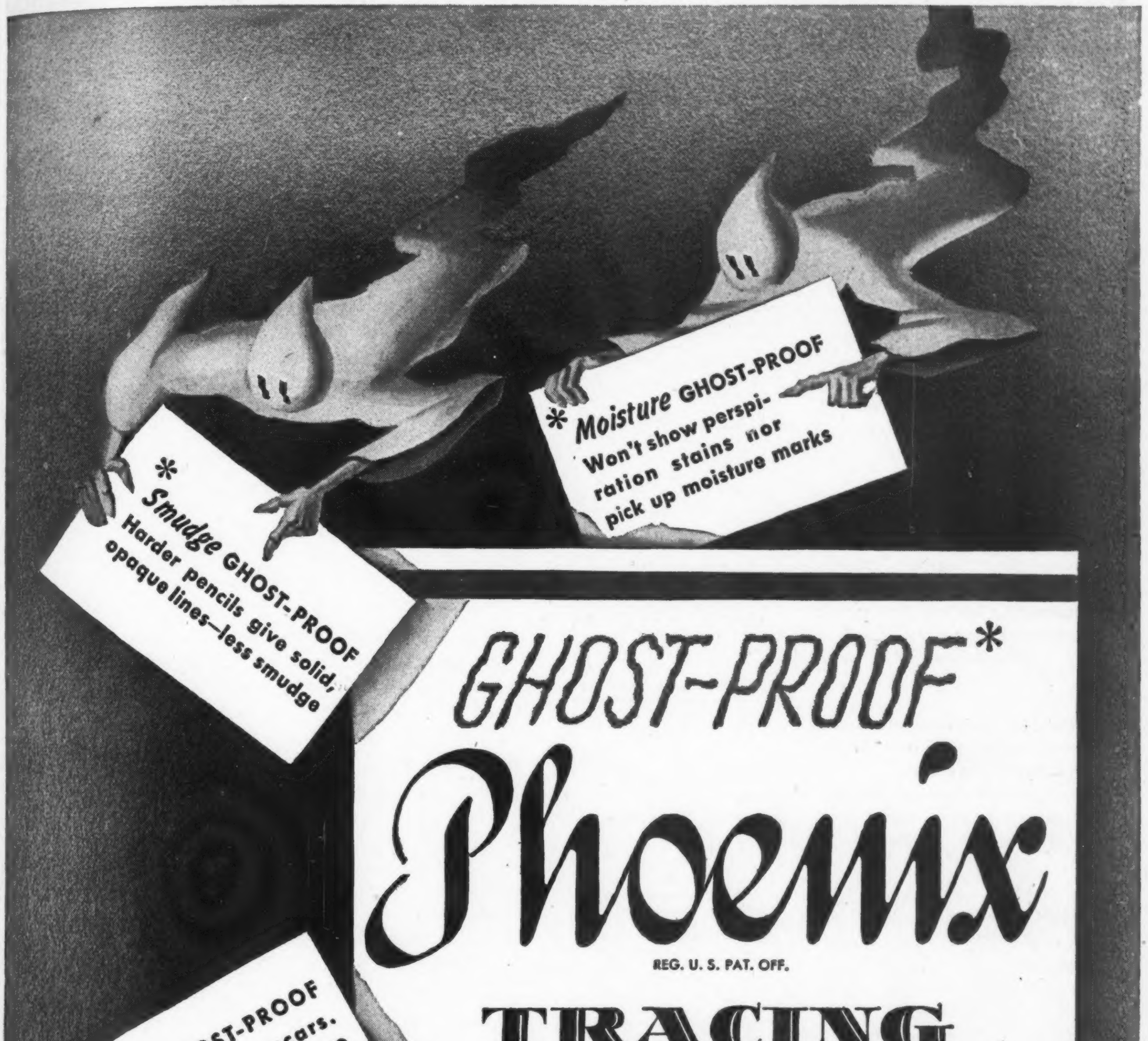
With headquarters at Bloomfield, N. J., Adolph Frankel has been named manager of the electronic tube sales department, Lamp division, Westinghouse Electric & Mfg. Co.

♦

Allen-Bradley Co. has announced the appointment of Rietze & Co., 1017 East Broadway, Louisville 4, Ky., as sales representative for the southern Indiana and western Kentucky territories.

♦

Establishment of a separate Texrope department has been announced by Allis-Chalmers Mfg. Co., Milwaukee. Position of manager and chief engineer of the new department has been


* Smudge GHOST-PROOF
Harder pencils give solid, opaque lines—less smudge
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Won't show perspiration stains nor pick up moisture marks
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given to T. C. Knudsen. Before reorganization the multiple V-belt power transmission section had been a part of the milling machinery department.

♦

Robert R. Hirsch has been appointed director of sales, Bunting Brass & Bronze Co., Toledo, O., succeeding George H. Adams who has been made executive vice president. Mr. Hirsch had been connected with SKF Industries Inc. for the past twenty-five years.

♦

Associated with the company since 1931, B. J. Brugge has been named welding engineer in the Detroit office of Lincoln Electric Co., Cleveland. Mr. Brugge will be available to firms needing assistance in the design and fabrication of present and postwar products.

♦

In order to consolidate manufacturing and distribution facilities in the Detroit district, Allegheny Ludlum Steel Corp., Brackenridge, Pa., has established new regional managership under H. N. Arbuthnot. Prior to his appointment Mr. Arbuthnot had been assistant general manager of sales at the main office in Brackenridge.

♦

National Forge & Ordnance Co., Irvine, Warren County, Pa., has been awarded the fourth White Star for sustained excellence in production.

♦

Lawrence S. Martz has been appointed assistant to the president, Micromatic Hone Corp., Detroit.

♦

According to a recent announcement by Firestone Tire & Rubber Co., H. M. Taylor has been appointed manager of the manufacturers' sales division at Akron. His former position as manager of the Detroit area has been assigned to D. J. Hutchins.

♦

Recently made vice president of Bendix Aviation Corp., Palmer Nicholls has announced that Bendix Aviation, Ltd., North Hollywood, Calif., is now operating as the Pacific division of the corporation.

♦

Purchase of International Metal Hose Co. has been announced by Gabriel Co., Cleveland. William H. Miller will continue as general manager of the new Metal Hose division, no change in personnel or policy having resulted from the purchase.

♦

Previously secretary of Seamless Steel Tube institute, Carl L. Zak has been named manager of tubular sales for Pittsburgh Steel Co.

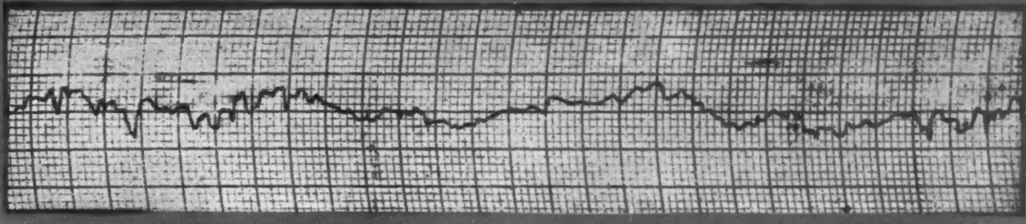
♦

Sheffield Steel Corp., Kansas City, Mo., has announced the appointment of C. A. Young as manager of sales, hot-rolled products division. Mr. Young has been connected with the company for ten years.

♦

Purchase of the Pittsburgh steel warehouse plant of Bethlehem Steel Co. has been announced by Joseph T. Ryerson & Son, Inc. Located in Carnegie, Pa., the plant is well situated for distribution in the Pittsburgh industrial area as well as for

Proof of a Better Finish



Each small square represents 1.0 micro inches. Micro inch r.m.s. = 0.9 — 1.6.

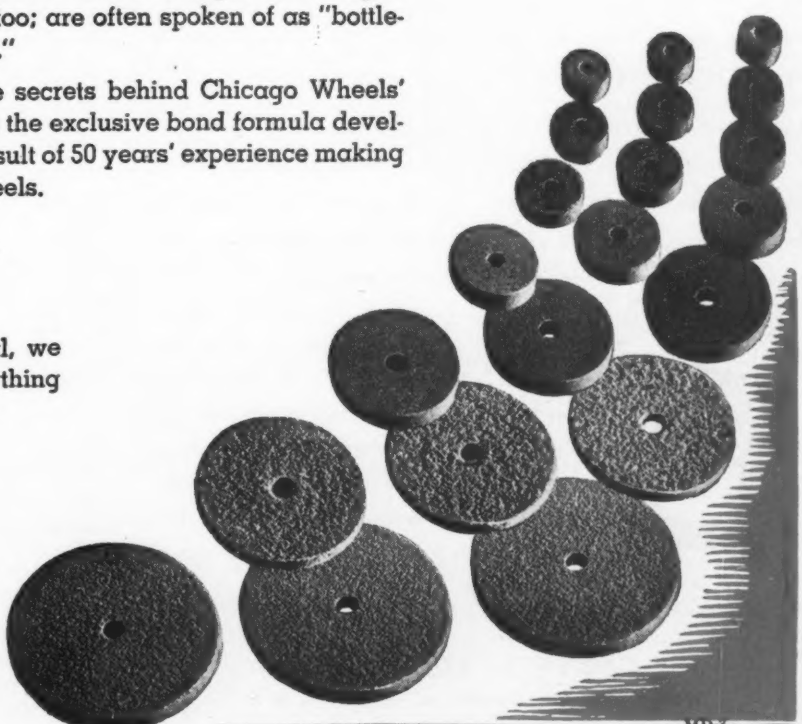
Surface Analyzer Tapes Show You Get A Better Finish With Chicago Wheels

These results were obtained at a rate of 10 pieces per hour in an aircraft parts plant. Material, X-13-15, Rockwell 60 to 57, grinds out .006 to .007 stock. Chicago Wheel used, ½ x ½ x ⅛", Grain 180, Grade L Arcite. Spindle Speed 40,000 r.p.m. Lapping and super finishing eliminated on this job.

Can you match that finish? Sounds phenomenal, but you can do the same thing with Chicago Wheels.

Squint your eye along the surface, test it by "feel" or be scientific and use a surface analyzer to measure your finish in micro inches — you'll find that Chicago Wheels give you better finishes, hold closer tolerances and have longer life. They're mighty fast, too; are often spoken of as "bottle-neck busters."

One of the secrets behind Chicago Wheels' superiority is the exclusive bond formula developed, as a result of 50 years' experience making grinding wheels.



Now Featuring Wheels Up to 3" in Diameter

For the duration, with full WPB approval, we are specializing on the small sizes — anything up to 3" in diameter.

Write for Catalog and we will include Engineering Survey Forms helpful in getting the right solution to your own grinding problems.

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trucking and freight service. W. W. Kopf, formerly of the Philadelphia plant, will be in charge of the plant office in Carnegie, while Howard L. Robinson of Cleveland will serve as Pittsburgh representative in the newly opened sales office in the Grant building, 330 Grant street, Pittsburgh 19.

♦

Promotion of Wade E. Canfield from division manager to assistant to the president has been announced by S. K. Wellman Co., Cleveland.

♦

With headquarters at 1329 North Highland Ave., Atlanta, Ga., Harris B. Carlock has been chosen to represent H. M. Harper Co. in Tennessee, Mississippi, Alabama, Georgia, North Carolina and South Carolina. Mr. Carlock will handle the company's line of nonferrous and stainless bolts, nuts, screws, washers and other fastenings.

♦

Connected with Bridgeport Brass Co. for the past eleven years, G. H. Tobelman has joined Wolverine Tube Division, Calumet & Hecla Consolidated Copper Co., as manager of the eastern territory with offices in New York City. Mr. Tobelman will supervise the other eastern branches in which the following changes have been made: Formerly in the New York area, C. E. Rinaman will manage the Boston territory; management of the Philadelphia territory will be under W. C. Gernhart, previously in Detroit; F. R. Meier will cover greater New York and A. S. Kingerley will remain in New Jersey.

♦

Construction of a new factory in Chicago has been started by Allied Control Co. of New York in order to expand manufacturing facilities for their relays, electrical control devices and fiber lock nuts.

♦

Recent announcement has been made of the appointment of Charles L. Foley as engineering sales representative for Kropp Forge Co., Chicago. Mr. Foley's territory will cover the metropolitan area of Manhattan, Staten Island, Long Island, twenty-five upstate counties, Eastern Pennsylvania including thirty-five counties, all of Maryland except Garrett and Allegany counties, Delaware, New Jersey and Connecticut. Since Connecticut has been taken out of the New England group and assigned to New York, W. R. Moore of West Hartford, Conn., is no longer New England representative of Kropp Forge Co.

♦

With the company since 1940, Alfred A. Lawrence has been named manager of the New England sales office of Dow Chemical Co. The office is located at 20 Providence street, Boston.

♦

Promotion of Whitley B. Moore from general manager of sales, Timken Steel and Tube division, to director of sales for all divisions has been announced by Timken Roller Bearing Co., Canton 6, O. C. H. McCollam, formerly assistant general sales manager of the Steel and Tube division has succeeded Mr. Moore.

♦

Opening of a Chicago sales office in the Pittsfield building, Suite 834, 55 East Washington street, has been announced by L & R Mfg. Co., Arlington, N. J. In charge of the new office will be John W. Denniston, western sales manager, who will

PERFORMANCE

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cover Minnesota, Iowa, Missouri, Kentucky, Wisconsin, Illinois, Michigan, West Virginia, Indiana, Ohio, western Pennsylvania and Tennessee. Prior to this appointment Mr. Denniston had been in charge of sales for the New England area.

For the past ten months acting sales manager, Vincent K. Alexander has been appointed sales manager of Manheim Mfg. & Belting Co., Manheim, Pa. Previously Mr. Alexander had been in charge of the Chicago office.

MEETINGS AND EXPOSITIONS

March 26-28—
American Society of Tool Engineers. Annual meeting to be held at Bellevue Stratford hotel, Philadelphia. Headquarters are at 2567 West Grand boulevard, Detroit.

April 1-3—
American Society of Mechanical Engineers. Spring meeting to be held at Birmingham, Ala. Additional information may be obtained from headquarters at 29 West Thirty-ninth street, New York. C. E. Davies is secretary.

April 2-5—
American Ceramic society. Second war congress and forty-sixth annual meeting to be held at Hotel William Penn, Pittsburgh. Ross C. Purdy, 2525 North High street, Columbus, O., is general secretary.

April 3-5—
American Society of Mechanical Engineers. Spring meeting to be held at Hotel Tutwiler, Birmingham, Ala. C. E. Davies, 29 West Thirty-ninth street, New York, is secretary.

April 5-7—
Society of Automotive Engineers. National aeronautical meeting to be held at Hotel New Yorker, New York. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary.

April 12-15—
Electrochemical society. Spring meeting to be held at Hotel Pfister, Milwaukee. Colin G. Fink, 3000 Broadway, New York, is secretary.

April 24-27—
National Electrical Manufacturers association. Spring meeting to be held at Palmer House, Chicago. W. J. Donald, 155 East Forty-fourth street, New York, is managing director.

April 25-28—
American Foundrymen's association. Third war production foundry congress and annual meeting to be held in Buffalo. An exhibit will also be held at Memorial auditorium. R. E. Kennedy, 22 West Adams street, Chicago, is secretary.

May 11-12—
Society of the Plastics Industry. Annual meeting to be held at Edgewater Beach Hotel, Chicago. W. T. Cruse, 295 Madison avenue, New York 17, is executive vice president.

May 22-24—
American Gear Manufacturers association. Twenty-eighth annual meeting to be held at Westchester Country club, Rye, N. Y. Newbold C. Goin, 301-2 Empire building, Pittsburgh, is manager-secretary.